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POWER, HEATING AND VENTILATION

PART I

STEAM POWER PLANTS

A Treatise for Designing and Constructing Engineers, Architects and Students

BY

CHARLES L. HUBBARD, B.S., M.E.

Consulting Engineer

SECOND EDITION

Fully Revised and Reset

McGRAW-HILL BOOK COMPANY, INC.

239 WEST 39TH STREET, NEW YORK

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PREFACE

This volume includes the first part of a treatise covering the design, construction and management of power, heating and ventilating plants. For the convenience of the reader the work has been divided into sections, and the chapters so arranged as to form three volumes, each complete in itself.

The matter contained in the following pages is intended to fully cover the design of small and medium size steam power plants.

The fundamental elements of the subject are first considered in some detail, after which the various pieces of apparatus employed in power work are fully described.

The later chapters are devoted to the selection of equipment for different conditions and the assembling of the same in the completed plant.

No attempt has been made to cover central station work; only plants similiar to those found in manufactories, office buildings and institutions being considered.

While care has been taken to treat the subject in a simple and concise manner, sufficient space has been devoted to the derivation of formulae and to the principles involved in the various methods of procedure, to make the work of value both to the student and practicing engineer.

Special thanks are due to the editors of various magazines for permission to use material which has appeared over the author's name in their publications.

As many of the chapters have been rewritten from several published articles upon similar subjects, it has been impossible to give suitable credit in the text, and for this reason it has seemed best to give references by chapters, as follows:

American Electrician, Chapters 2, 3, 4, 6, 7, 8, 9, 10, 11, 12, 13, 14, 16.
Electrical World, Chapter 12.
Electrical Review, Chapter 16.
Engineering Review, Chapter 14.

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Machinery, Chapters 2, 6.
Power, Chapters 4, 10.
Science and Industry, Chapters 4, 12, 13.
Steam, Chapter 3.
Steam Engineering, Chapters 2, 7, 11, 12, 13, 15.
The Engineer, Chapters 1, 2, 3, 4, 5, 6, 7, 8, 9, 11, 13, 15.

Acknowledgment is also due for the courtesy extended by the authors and publishers of the following books of reference:

Steam Boilers, and Mechanical Engineers' Pocket Book, by Kent, data on heat, steam, boiler horse power, boiler design, boiler furnaces, chimneys, mechanical draft.
Steam Power Plants, by Meyer, data on chimneys and feed piping.
Mechanical Draft, B. F. Sturtevant Company, data on boiler horse power, chimneys, mechanical draft and fans.
Steam Boilers, by Peabody and Miller, data on boiler horse power, boiler design, corrosion and chimneys.
Steam Turbines, by French, data on steam turbines.

Thanks are also extended to the makers of various kinds of apparatus shown, for the use of tables and cuts.

It has been the intention of the author to give credit in every case where due, but if through any oversight this has not been done, it will be considered a favor to have such oversight reported in order that it may be corrected in future editions.

CHARLES L. HUBBARD.

Boston, Mass.

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PART I

STEAM POWER PLANTS

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STEAM POWER PLANTS

CHAPTER I.

HEAT

As the various effects of heat enter so largely into the subjects treated in the following pages, a brief discussion of its nature and some of its more important properties may be of assistance to the reader, by fixing clearly in his mind the exact meaning of the terms most frequently used.

Heat is recognized by the bodily sensation of touch, and by means of this we are able to judge roughly of the temperature of an object. The terms heat and cold are relative only, and when we speak of a body as being hot or cold we mean that its temperature is higher or lower than that of some other body with which it is compared.

Theory of Heat.—There have been many theories in regard to the exact nature of heat, but the accepted one at the present time is that of molecular vibration. It is thought that heat is a vibration of the molecules of which a body is composed, and that the degree of heat, or temperature, depends upon the velocity and amplitude of the vibrations.

Bodies in general expand when heated. This is explained by the fact that the molecules when in motion force each other apart, so that the whole body occupies a greater space than before.

Solids melt and liquids turn to gas when heated to certain temperatures. This means that when the vibrations become sufficiently violent the molecules are pushed apart to such an extent that their attraction for each other is greatly lessened and the character of the substance is changed through lack of stiffness or

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stability. Heat can be produced by hammering or distorting a body in any way. Hammering naturally shakes the molecules, and thus increases their agitation.

The weight of a body is not changed by increasing or decreasing its temperature. The movement of an object makes it neither lighter nor heavier, therefore the vibration of its molecules does not change their weight.

Heat will pass from one body to another when one is "hotter" than the other. This naturally follows because a body which is vibrating vigorously will give some of its motion to one having a less motion, when brought in contact with it.

From the above, we see that the present theory accounts for the various phenomena connected with heat, and further, has enabled people to make predictions which have been verified by experiment.

Temperature and Its Measurement

Temperature.—The intensity of the heat in a body, that is, the velocity of vibration of its molecules, is called the temperature. This, however, does not indicate the *quantity* of heat which it contains. For example, a small bar of iron may be heated to a white heat and still contain a much smaller quantity of heat than a larger bar at a lower temperature.

Thermometers.—The temperature of a body is determined by comparing it with some substance whose intensity of heat is known. The simplest way of doing this is to use a substance the volume of which changes a definite amount with the addition or removal of a given quantity of heat, and always has the same volume for the same degree of temperature. A substance having the above property is placed in a glass bulb, to which is connected a tube of small bore. As heat is applied to the bulb the substance expands into the tube. Marks are then made at the height at which the substance stands when it has a certain degree of heat. In order to make the comparison, we place the bulb of the tube, which is called a *thermometer,* in contact with the body whose temperature we wish to know, and then note the height of the substance within the tube. We say two bodies have the same temperature, when no heat will pass from one to the other if they are

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placed in contact. Therefore the reading of a thermometer indicates the temperature of the body with which it is in contact, but cannot give the *quantity* of heat it contains.

Mercury is commonly used in thermometers, although air, water, alcohol and other substances are often employed.

Fahrenheit Scale (*F.*).—The scale most frequently used in this country is known as the Fahrenheit scale. The graduations in this case are obtained by noting the position of the mercury column when the bulb is placed in melting ice, and again when placed in steam which is being evaporated under atmospheric pressure. This difference in the heights of the column is divided into 180 equal parts, called degrees. The freezing point of this scale is marked 32 degrees, which makes the boiling point 32+180=212 degrees above zero.

Centigrade Scale (*C.*).—In the French system the Centigrade thermometer is used. The freezing point on this scale is marked zero, and the boiling point 100.

Methods of Conversion.—As the difference between the freezing and boiling points on the Fahrenheit scale is divided into 180 degrees, and on the Centigrade scale into 100 degrees, it follows that 1 degree C.=180÷100=1.8 degree F., and 1 degree F.=100÷180=0.55 degree C.

Zero on the Centigrade scale is at the freezing point, and on the Fahrenheit scale it is 32 degrees below the freezing point. Therefore, to change the Fahrenheit scale to Centigrade, we must first subtract 32 and then multiply the remainder by 0.55.

The methods of conversion may be represented by the following equations, in which C. is the reading in degrees Centigrade and F. the reading in degrees Fahrenheit.

$$C.=0.55\times(F.-32)$$
$$F.=(1.8\times C.)+32$$

Example.—Reduce 60° F. to the Centigrade scale. (60°—32) ×0.55=15.4°C. The reason for this is evident because 60° F. is 60—32=28° above the freezing point and 1° F.=0.55° C.; then 28° F.=28×0.55=15.4° C.

In like manner readings on the Centigrade scale can be changed to the Fahrenheit scale by first multiplying by 1.8 and then adding 32 to the product.

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Absolute Temperature.—The volume of a "perfect gas" at a temperature of zero Centigrade varies $^1/_{273}$ of its volume for a change of one degree in its temperature, when under constant pressure. Assuming this to hold true for other temperatures, the volume of a perfect gas will become zero, or ***nothing,*** when cooled to a temperature of 273° below zero. This point is called the ***absolute zero*** and corresponds to 460.66 degrees below zero on the Fahrenheit scale.

To reduce Centigrade readings to absolute, add 273.

To reduce Fahrenheit readings to absolute, add 460.66, or 461 in round numbers.

Example.—Plus 10° F.$=(461+10)=471°$ absolute, and minus 10° F.$=(461-10)=451°$ absolute. The absolute zero is only an imaginary condition, but absolute temperatures are often used in making computations where ratios occur.

General Definitions for the English System

Unit of Heat.—The unit of heat is taken as the quantity required to raise the temperature of one pound of pure water one degree at its point of greatest density, which is at about 39° F.

The quantity of heat required to raise one pound of water through successive degrees is not quite constant, but increases slightly as the temperature rises. For all practical purposes, however, we may neglect this, and define a heat unit, or British Thermal Unit (B. T. U.) as the quantity of heat required to raise the temperature of one pound of water one degree Fahrenheit.

Latent Heat.—When heat is applied to a body and it passes from a solid to a liquid state, its temperature remains practically constant during the process. The heat which disappears during this change is called the *latent heat of fusion,* and represents the work done in tearing apart the molecules of the substance. When the body solidifies, this heat is given back again. The heat required to change a liquid into a gas is called the *latent heat of evaporation,* or *vaporization.* The heat necessary to melt ice into water, and then evaporate the water into steam, are familiar examples of the above.

Specific Heat.—The quantity of heat required to raise the tem-

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perature of a body one degree is called its *thermal capacity*. The thermal capacity of a pound of pure water is one B. T. U. and is greater than that of any other known substance. The thermal capacity of a given weight of any other substance, compared with an equal weight of water, is called its *specific heat*, and is always less than 1. We may thus define specific heat as the quantity of heat, expressed in thermal units, necessary to raise the temperature of one pound of any substance one degree.

Conduction.—The transfer of heat between two bodies which are in contact is called *conduction*. When this transfer takes place between different parts of one continuous body, it is called *internal* conduction. When it passes from one body to another it is known as *external* conduction. If one end of an iron bar be held in the fire, the other end soon becomes heated by conduction. If a piece of wood were used in place of the bar, the end in the fire would burn, and but little heat would be transferred to the other end. Thus we see that some substances are good conductors of heat while others are not.

Convection.—We know from experience that lighter solids or liquids will float upon a heavier liquid. When a portion of a liquid is heated and expands, it rises to the surface because it is lighter, or rather because the cooler liquid surrounding it falls to the bottom of the vessel and forces the lighter liquid upward. Currents set up in this manner are called *convection* currents. Convection also takes place in gases.

Radiation.—We have learned that in a heated body the molecules are supposed to be in violent agitation. The motion of these vibrating molecules or particles is communicated to the surrounding ether, and transmitted by it with great rapidity in the form of waves. Heat transmitted in this manner is called *radiant* heat, and the process is called *radiation*.

Radiant heat travels in straight lines through a uniform medium, and the radiation is equal in all directions. The heat from a fireplace or steam radiator is radiant heat, and passes through the air to the walls of a room which absorb it. As these become warm the heat is given off to the surrounding air by conduction and is diffused throughout the room by convection. A certain amount of heat is also absorbed by the air directly from the radiator in the same way.

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Thermodynamics.—That branch of science which considers the relation between heat and mechanical work is called *thermodynamics*. It deals with the numerical relation between the quantity of heat used and the amount of work done.

We have already seen that heat is produced by hammering or distorting a body. Other examples of the generation of heat by work are found in the bearings of machinery, cutting tools, etc.

In the steam engine hot steam is admitted to the cylinder, it does work on the piston, and is discharged cooler than it entered; heat has disappeared and work has been done. It is plain that when work is done *on* a body heat is generated in it, and when work is done *by* a body some of its heat disappears. Experiment shows that when a certain amount of work has been done by a hot body a corresponding amount of heat has disappeared.

This subject has been investigated with great care, and it has been found that the ratio of the unit of heat to the unit of work is constant. That is, if 778 foot-pounds of work are done on a body, enough heat may be generated to raise the temperature of one pound of water one degree F. This equals one B. T. U.

Conversely, if a hot body—steam, for example—does work, one B. T. U. will disappear for every 778 foot-pounds of work done. This number, 778 is called the ***heat equivalent of work,*** or the *mechanical equivalent of heat.*

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CHAPTER II.

STEAM

Although a body cannot become hotter unless heat passes into it or is generated in it, heat may pass into some bodies under certain conditions without raising their temperatures. The making of steam is an example of this and may be illustrated as follows: Let a vertical cylinder be fitted with a piston which is free to move up and down, but which may be loaded as much as desired. Suppose there is a certain quantity of water,—one pound for instance,—in the bottom of the cylinder and that the piston rests upon its surface. If heat be applied to the cylinder by means of a gas flame or a fire, certain results will be noticed.

First, the temperature of the water will gradually rise, until it reaches a point at which steam begins to form. Second, while steam is being formed the piston will rise, exerting a steady and constant pressure; but, although the fire continues as hot as ever, the temperature of the water and steam will remain constant. Steam formed in this way in the presence of water is called *saturated steam.* This state will continue until all of the water has been evaporated, the temperature remaining the same. Third, after the water has been evaporated, the piston will still continue to rise, but the temperature, which remained constant while steam was forming, will increase, and the steam is now said to be *superheated.*

The properties of steam with which the engineer has most to do are given in Table I., called a *steam table.*

If there had been a different load on the piston, so that the pressure on the steam and water beneath it had been different, the temperature at which steam was formed would have varied from that noted in the above case. If the pressure had been greater, the temperature would have been higher; if the water had been open to the atmosphere, steam would have formed when the water reached a temperature of 212° F. It has been shown by experiment that there is a definite relation between the temperature and the pressure exerted by saturated steam; that is, for

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every pressure there is a corresponding and fixed temperature. There is no such direct relation between the temperature and pressure of superheated steam. Superheated steam at 60 pounds gauge pressure may have any temperature above 307°. At a stated pressure the temperature and volume of a given weight of superheated steam are always greater than the temperature and volume of the same weight of saturated steam.

Steam in contact with water is always *saturated.* If water is present in the form of a mist or spray, the steam is said to be *wet.* If there is no water present in this form it is called *dry.* From the results of the above experiment the following brief definitions may be given:—

Saturated Steam is steam of the temperature due to its pressure, not superheated.

Superheated Steam is steam heated to a temperature above that due to its pressure.

Dry Steam is steam which contains no moisture. It may be either saturated or superheated.

Wet Steam is steam containing moisture in the form of a mist or spray. It has the same temperature as dry saturated steam of the same pressure.

Water introduced into the presence of superheated steam will flash into a vapor until the temperature of the steam is reduced to that due to its pressure.

Water in the presence of saturated steam has the same temperature as the steam.

Should cold water be introduced, lowering the temperature of the whole mass, some of the steam would be condensed, reducing the pressure and temperature of the remainder until an equilibrium was established.

Returning to the illustration of the cylinder, it will be remembered that while steam was forming, the temperature of both the water and steam remained constant, although more heat was being supplied all of the time. This heat, which apparently disappears in the formation of steam, has been defined as *latent heat,* and when the steam is condensed it is again given out. The quantity of heat which becomes latent in changing water into steam decreases. as the temperature at which the steam is formed, increases.

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Steam Tables

The properties of steam most frequently employed in engineering work are given in Table I., called a *steam table.*

Refering to this we find the following headings to the different columns:—

1. Pressure in pounds per square inch.
2. Temperature in degrees F.
3. Heat in the liquid above 32° F.
4. Internal latent heat.
5. External latent heat.
6. Latent heat of evaporation.
7. Total heat of evaporation.
8. Weight of a cubic foot of steam, in pounds.
9. Volume of a pound of steam, in cubic feet.
10. Ratio of volume of steam to volume of equal weight of water.

Now let us take these up briefly in order and see what each means.

1. The pressure shown by a steam gauge is 14.7 pounds less than the actual pressure inside the boiler, because the pressure of the atmosphere is acting upon the outside of the gauge, and the pointer indicates only the difference between the internal pressure and that of the atmosphere. For this reason 14.7 pounds, or in whole numbers, 15 pounds, must be added to the gauge reading to get the absolute pressure within the boiler. All comparisons in which ratios occur should be in absolute pressures. Some steam tables give the absolute pressure, while others give gauge readings.

2. As already stated, saturated steam under a given pressure always has the same temperature. The temperatures corresponding to different pressures have been accurately determined and recorded, so that by knowing the pressure one can find the corresponding temperature at once from a steam table; or, knowing the temperature, the pressure may be found in the same manner. Columns 3, 4, and 5 may best be explained by reference to the illustration of the cylinder.

3. This shows the number of B. T. U. which have been

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TABLE I.

THE PROPERTIES OF SATURATED STEAM.

1	2	3	4	5	6	7	8	9	10
Pressure in pounds per square inch.	Temperature in degrees F.	Heat in the liquid above 32 degrees.	Internal latent heat.	External latent heat.	Latent heat of evaporation.	Total heat of evaporation.	Weight of a cubic foot of steam in pounds.	Volume of a pound of steam in cubic feet.	Ratio of volume of steam to volume of equal weight of water.
ABSOLUTE PRESSURE.									
1	102.0	70.0	961.4	61.6	1043.0	1113.0	.0080	330.4	20632
2	126.3	94.3	962.0	64.1	1026.1	1120.4	.0058	171.9	10730
3	141.6	109.7	950.0	65.6	1015.4	1125.1	.0085	117.3	7325
4	153.1	121.3	940.6	66.8	1007.4	1128.6	.0112	89.51	5588
5	162.3	130.5	933.2	67.6	1000.9	1131.4	.0138	72.56	4530
6	170.1	138.4	927.0	68.4	995.4	1133.8	.0163	61.14	3816
7	176.9	145.2	921.6	690.	990.7	1135.9	.0189	52.89	3302
8	182.9	151.2	916.9	69.6	986.5	1137.7	.0214	46.65	2912
9	188.3	156.8	912.6	70.1	982.7	1139.4	.0239	41.77	2607
10	193.3	161.6	908.7	70.5	979.2	1140.9	.0264	37.83	2361
11	197.8	166.2	905.1	70.9	976.0	1142.3	.0289	34.59	2159
12	202.0	170.4	901.7	71.3	973.1	1143.5	.0314	31.87	1990
13	205.9	174.4	898.7	71.6	970.3	1144.7	.0338	29.56	1845
14	209.6	178.1	895.8	72.0	967.7	1145.8	.0362	27.58	1721
GAUGE PRESSURE.									
0	212.0	180.5	893.9	72.17	966.0	1146.6	.0379	26.37	1646
1	216.3	184.9	890.4	72.55	963.0	1147.9	.0411	24.33	1519
2	219.4	188.0	888.0	72.81	960.8	1148.8	.0435	22.98	1434
3	222.4	191.0	885.6	73.06	958.7	1149.7	.0459	21.78	1359
4	225.2	193.9	883.4	73.30	956.7	1150.6	.0483	20.70	1292
5	227.9	196.6	881.3	73.52	954.8	1151.4	.0506	19.73	1231
6	230.5	199.2	879.2	73.74	952.9	1152.2	.0530	18.84	1176
7	233.0	201.8	877.2	73.94	951.2	1153.0	.0554	18.04	1126
8	235.4	204.2	875.3	74.13	949.5	1153.7	.0578	17.30	1080
9	237.8	206.6	873.5	74.32	947.8	1154.4	.0601	16.62	1038
10	240.0	208.8	871.7	74.50	946.2	1155.1	.0625	16.00	998.4
12	244.3	213.2	868.4	74.85	943.2	1156.4	.0672	14.88	928.8
14	248.3	217.3	865.2	75.17	940.3	1157.6	.0718	13.91	868.5
16	252.1	221.1	862.2	75.46	937.6	1158.8	.0765	13.07	815.8
18	255.7	224.8	859.4	75.74	935.1	1159.9	.0811	12.32	769.2
20	259.2	228.3	856.7	76.00	932.6	1161.0	.0857	11.66	727.9
22	262.5	231.6	854.1	76.25	930.3	1162.0	.0903	11.07	690.8
24	265.6	234.8	851.6	76.49	928.1	1162.9	.0949	10.53	657.5
26	268.6	237.9	849.2	76.72	925.9	1163.8	.0995	10.05	627.3
28	271.5	240.8	847.0	76.93	923.9	1164.7	.1040	9.60	599.9
30	274.3	243.6	844.7	77.13	921.9	1165.6	.1086	9.20	574.7
32	277.0	246.4	842.7	77.33	920.0	1166.4	.1131	8.83	551.7
34	279.6	249.0	840.6	77.51	918.1	1167.2	.1176	8.49	530.5
36	282.1	251.6	838.7	77.69	916.3	1167.9	.1221	8.18	510.9
38	284.5	254.1	836.7	77.87	914.6	1168.7	.1266	7.89	492.8
40	286.9	256.5	834.9	78.03	912.9	1169.4	.1311	7.62	475.9
42	289.2	258.8	833.1	78.19	911.3	1170.1	.1356	7.37	460.2
44	291.4	261.1	831.3	78.35	909.7	1170.8	.1401	7.13	445.5
46	293.6	263.3	829.6	78.49	908.1	1171.5	.1445	6.91	431.7
48	295.7	265.5	828.0	78.64	906.6	1172.1	.1490	6.70	418.8
50	297.8	267.6	826.4	78.78	905.1	1172.7	.1535	6.51	406.6
52	299.8	269.6	824.8	78.91	903.7	1173.3	.1579	6.33	395.2
54	301.8	271.6	823.3	79.04	902.3	1173.9	.1623	6.15	384.5
56	303.7	273.6	821.8	79.17	900.9	1174.5	.1667	5.99	374.3
58	305.6	275.5	820.3	79.29	899.6	1175.1	.1712	5.84	364.6
60	307.4	277.4	818.9	79.41	898.2	1175.7	.1756	5.69	355.5
62	309.2	279.2	817.4	79.52	896.9	1176.2	.1800	5.55	346.8
64	311.0	281.0	816.1	79.64	895.7	1176.7	.1844	5.42	338.5

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TABLE I.

THE PROPERTIES OF SATURATED STEAM. (Continued.)

1	2	3	4	5	6	7	8	9	10
Pressure in pounds per square inch.	Temperature in degrees F.	Heat in the liquid above 32 degrees.	Internal latent heat.	External latent heat.	Latent heat of evaporation.	Total heat of evaporation.	Weight of a cubic foot of steam in pounds.	Volume of a pound of steam in cubic feet.	Ratio of volume of steam to volume of equal weight of water.
GAUGE PRESSURE.									
66	312.7	282.8	814.7	79.75	894.4	1177.3	.1898	5.29	330.6
68	314.4	284.5	813.4	79.85	893.2	1177.8	.1932	5.17	323.1
70	316.0	286.2	812.1	79.96	892.0	1178.3	.1975	5.06	315.9
72	317.7	287.9	810.8	80.06	890.9	1178.8	.2019	4.95	309.1
74	319.3	289.5	809.6	80.16	889.7	1179.3	.2063	4.84	302.5
76	320.8	291.1	808.4	80.26	888.6	1179.8	.2107	4.74	296.3
78	322.4	292.7	807.2	80.35	887.5	1180.2	.2150	4.65	290.2
80	323.9	294.3	804.0	80.44	886.4	1180.7	.2194	4.55	284.5
82	325.4	295.8	804.8	80.53	885.3	1181.1	.2237	4.46	279.0
84	326.9	297.3	803.7	80.62	884.2	1181.6	.2281	4.38	273.7
86	328.3	298.8	802.5	80.71	883.2	1182.0	.2324	4.30	268.5
88	329.7	300.2	801.4	80.79	882.2	1182.5	.2368	4.22	263.6
90	331.1	301.7	800.3	80.87	881.2	1182.9	.2411	4.14	258.9
92	332.5	303.1	799.2	80.95	880.2	1183.3	.2454	4.07	254.3
94	333.9	304.5	798.2	81.03	879.2	1183.7	.2497	4.00	249.9
96	335.2	305.9	797.1	81.10	878.2	1184.1	.2541	3.93	245.7
98	336.5	307.2	796.1	81.18	877.3	1184.5	.2584	3.87	241.6
100	337.8	308.6	795.1	81.26	876.3	1184.9	.2627	3.80	237.6
105	341.0	311.8	792.6	81.44	874.0	1185.9	.2735	3.65	228.3
110	344.1	315.0	790.2	81.61	871.8	1186.8	.2842	3.51	219.6
115	347.1	318.1	787.9	81.77	869.6	1187.8	.2949	3.39	211.6
120	350.0	321.1	785.6	81.93	867.5	1188.6	.3056	3.27	204.2
125	352.8	324.0	783.5	82.08	865.5	1189.5	.3163	3.16	197.3
130	355.5	326.8	781.3	82.22	863.5	1190.3	.3269	3.05	190.9
135	358.2	329.5	779.3	82.36	861.6	1191.2	.3376	2.96	184.9
145	363.3	334.8	775.3	82.61	857.9	1192.7	.3588	2.78	173.9
155	368.2	339.8	771.5	82.85	854.3	1194.2	.3800	2.63	164.3
165	372.8	344.7	767.9	83.07	850.9	1195.6	.4012	2.49	155.6
175	377.3	349.3	764.4	83.27	847.7	1197.0	.4222	2.36	147.8
185	381.6	353.7	761.1	83.46	844.5	1198.3	.4433	2.25	140.8
195	385.7	358.0	757.9	83.64	841.5	1199.5	.4642	2.15	134.5
205	389.7	362.1	754.8	83.80	838.6	1200.8	.4852	2.06	128.7

used in raising the temperature of the water from 32° F. to that at which steam is formed, for any given pressure.

4. The internal latent heat is that which has been expended in work, in the process of evaporation, that is, in the tearing apart of the particles of water against molecular attraction.

5. The external latent heat is used in the work of expansion, and represents the work required to raise the piston and weight in the illustration of the cylinder.

6. The latent heat of evaporation is the sum of the internal and external latent heats, and would again be given out should the

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steam be condensed. This is sometimes called the latent heat of steam or heat of vaporization.

7. The total heat of evaporation is the whole quantity of heat required to raise the water from 32° F. and evaporate it into steam at the given pressure, and is equal to the heat in the liquid plus the internal and external latent heat.

8. This quantity, called the *density,* shows the weight in pounds of one cubic foot of steam at the pressure taken.

9. This column gives the volume of one pound of steam at the given pressure, and is called the *specific volume.*

10. The quantities given in this column are sufficiently explained in the table, and require no further explanation.

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CHAPTER III.

FUELS AND COMBUSTION

Steam for all commercial purposes is generated by the heat given off during the combustion of fuel, hence, a thorough understanding of the characteristics and heat values of different fuels is necessarry in the efficient design and management of all power plants where steam is employed.

Fuels

Although coal is by far the most important fuel in use at the present time, wood, peat, oil, gas, and artificial fuels, such as coke and briquetts are used to a considerable extent in different localities and under varying conditions.

Combustibles.—The heat producing properties of all fuels are called combustibles, and, in natural fuels, are found in combination with various impurities or non-combustibles. The most important constituent of all fuels is carbon, while others of a volatile nature, such as hydrogen, hydro-carbon compounds, and sulphur, are found in varying quantities in different fuels. Carbon is found in both the free and combined states, but is not associated in the former condition with any combustible element.

Hydrogen is a gas and usually occurs in combination with carbon or oxygen. When combined with the latter it forms water, which is non-combustible.

The hydro-carbons, so called, consist of various combinations of hydrogen and carbon, such as marsh gas, olefiant gas, pitch, tar, etc. Sulphur occurs only in small quantities and is usually combined with iron. It is an undesirable constituent, as it produces a corrosive effect upon the plates of the boiler and furnace.

Non-Combustibles.—The most important of the non-combustibles are ash, water, oxygen, and nitrogen.

Ash includes all solid residue which remains after the combustion of fuel.

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Water occurs as moisture in its natural state, uncombined with other constituents.

Oxygen is found in a free state, but more commonly in combination with hydrogen, as water.

Nitrogen occurs only in small quantities, and in the free state.

Coal

As previously noted, coal is the most important fuel employed for the generation of steam. This is divided by the common market classification into anthracite, semi-anthracite, semi-bituminous, bituminous and lignite.

Anthracite Coal.—This is the nearest to pure carbon and represents the highest quality of fuel known. It is used almost entirely for domestic purposes, and for power where entire freedom from smoke is required. This advantage, however, is much less than formerly, owing to the improvements in furnaces for burning bituminous coal, and a better knowledge of the methods of operating them. As the amount of ash varies to a considerable extent in different coals of the same class, it is customary to give the heating value per pound of combustible rather than per pound of coal. Kent's table gives the heat value of anthracite as running from 14,600 to 14,800 B. T. U. per pound of combustible.

Semi-Anthracite Coal.—This is similiar to anthracite in most respects, but has a higher percentage of ash and slightly more oxygen. The supply of this grade, however, is so small that it is not used to any great extent in the operation of power plants.

Semi-Bituminous Coal.—This coal is one of the finest varieties for use in boiler furnaces, having a high heat value and producing less smoke than bituminous. In apperance it is much like anthracite but contains more volatile matter. As the supply is limited, this coal is not widely used for power purposes except in the East. The heating value per pound of combustible is given as 15,500 to 16,000 B. T. U.

Bituminous Coal.—This includes both the caking and non-caking varieties, which have distinctly different characteristics. Non-caking coal is rather hard and brittle; it burns with a smoky yellow flame and does not melt or fuse when heated in the furnace.

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Caking coal, on the other hand, fuses and swells into a mass which it is necessary to break up from time to time to allow the gases to pass through. This variety is rich in volatile matter, and produces a good deal of smoke which is difficult to avoid. Bituminous coal is usually a dense black, with a resinous luster, and the best quality is soft and silky to the touch. The heat value per pound of combustible for the eastern varieties runs from 14,800 to 15,200 B. T. U., and for the western varieties from 13,500 to 14,800 B. T. U.

Lignite.—This varies in color from brown to a deep black, according to quality; the darker the color the more valuable for fuel purposes. The amount of ash varies so greatly that some lignites are worthless for the generation of steam. They have a low heat value and require a strong draft to secure the best results when burned under a boiler. The heat value per pound of combustible is given as 11,000 to 13,500 B. T. U.

Other Fuels

Coke.—This is a manufactured fuel, being produced by heating bituminous coal with an air supply sufficient to burn only the volatile hydrocarbons. In some cases the coal is heated in air-tight kilns to which external heat is applied, as in gas works.

Briquetts.—The forming of coal dust into briquetts with some form of binder is a method of utilizing a waste product, which is increasing to some extent in this country. At the present time however, the cost of manufacture seems to be such as to prevent the general use of this fuel for power plant purposes.

Liquid Fuel.—The use of oil as a fuel is becoming a matter of considerable importance in certain sections of the country, both for stationary boilers and locomotives. Various steamship lines have also equipped some of their boats for burning oil with very satisfactory results. There are two kinds of oil in general use for this purpose. One is the crude petroleum as it comes from the wells, and the other is the residuum which is left after the lighter oils have been distilled. The use of liquid fuel has thus far been confined principally to the oil producing regions, but numerous plants in other sections have been operated successfully in competition with coal.

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The principal advantages claimed for oil as a fuel as compared with coal are: The absence of soot and ashes, resulting in clean heating surfaces and increased steaming capacity of the boilers; a more even distribution of heat in the combustion chamber, as it is not necessary to open the feed doors to supply fresh fuel; a reduction in the cost of handling, as this is done mechanically or by gravity; absence of dust, ashes and clinkers and ease of regulating the fires.

Opposed to these advantages are: danger of explosion under certain conditions, if stored in too warm a place; higher cost in many localities; and excessive noise from the burners if not properly constructed and adjusted.

Oil Compared with Coal.—The heat produced by the combustion of 1 pound of petroleum will vary from about 17,000 to 20,000 heat units as against 12,00 to 14,000 for an equal weight of coal, making, on an average, the heating value of 1 pound of petroleum equivalent to approximately 1½ pounds of coal. In actual practice the ratio is found to be somewhat greater, and under ordinary conditions from 3½ to 4½ barrels (42 gallons per barrel) of oil are considered equivalent to 1 ton of coal (2,000 pounds).

Taking this as a basis the relative costs of the two fuels for any given locality can be approximately computed, due allowance being made for the reduction of labor in the case of oil. For a more accurate comparison a careful analysis of both the coal and oil should be made to determine their heating values.

Combustion

In order properly to design a furnace for the economical burning of any particular grade of fuel it is necessary to understand the principles of combustion. This information is important, not only to the designing engineer, but equally so to the operating engineer and those in charge of the fireroom.

Chemistry Involved in Combustion.—The principal elements in fuel which enter into the process of combustion are carbon, hydrogen, and oxygen. Combustion, in its broadest sense, is the chemical reaction between carbon and oxygen, forming carbonic

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acid gas, or carbon dioxide; and between hydrogen and oxygen, forming water vapor or steam.

While the atmospheric air is not an element, it is so closely connected with the process of combustion that it should be considered in this connection. Air is a mechanical mixture, made up of one part of oxygen to four of nitrogen, by volume. It also contains a small percentage of carbon dioxide, especially in the locality of large cities, and also a small amount of water vapor.

Nitrogen does not unite with the elements of the fuel during combustion and is therefore one of the chief losses in the operation of a boiler furnace. Its effect is that of diluting the oxygen and taking from the fire a considerable amount of heat. On the other hand, if it were not for this dilution, combustion of the fuel with pure oxygen would produce a heat far beyond the endurance of the boiler plates and tubes so that, after all, it serves a useful purpose in reducing the temperature to a point within practical limits.

Process of Combustion.—The actual process of combustion as it takes place in the boiler furnace is made up of the following stages:

First, the absorption of heat to the kindling point of the fuel; second, the distillation and burning of the volatile gases; and third, the combustion of the fixed carbon. In order that combustion may proceed properly, after once being started, it is necessary to have a sufficient, but not excessive supply of oxygen, and that it be brought into intimate physical contact with the combustable elements of the fuel.

During the first stage of combustion after fresh fuel has been thrown upon the fire, it continues to absorb heat until it reaches the kindling temperature, which varies somewhat with the constituents of the fuel.

The next stage is the distillation of the hydro-carbons which are driven off by the heat and burned by uniting with the oxygen of the air, which passes from the ash pit up through the bed of the fire.

The product of this combustion is carbon monoxide, carbon dioxide, and water vapor. Only the solid portion of the coal now

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remains, which is made up of the fixed carbon and the non-combustible ash.

If sufficient air is supplied, the oxygen unites with the carbon, forming carbon dioxide; if, however, the air supply is restricted, the products of combustion will consist of a mixture of carbon dioxide and carbon monoxide. After complete combustion has taken place, only ash remains in the furnace.

Air Supply for Combustion.—Theoretically, 11.52 pounds of air are necessary to burn 1 pound of pure carbon, which is approximately 12 pounds of air per pound of coal. In actual practice it is impossible to get a sufficiently intimate mixture of the coal and air to utilize all of the oxygen in the air supply, hence it is necessary, with a natural draft, to provide about 24 pounds of air per pound of coal, and 18 pounds with mechanical draft.

Smokeless Combustion.—Smokeless combustion, which is either being sought or enforced in many cities and towns, means an approach to the conditions of perfect combustion. The smoke given off by the chimneys of cities and large towns is not only unsightly, and injurious to both exterior and interior architectural decorations, but is unhealthful as well. While the waste of fuel in the smoke itself is not great, it indicates a condition of incomplete combustion and shows that invisible unconsumed gases are passing away with the smoke.

The principles of perfect combustion, while apparently simple, are not always easy to carry out, especially with those grades of bituminous coal which are high in volatile matter. It is during the first stages of combustion, while the hydro-carbons are being given off, that smoke is formed, there being but little difficulty in this direction after the bed of fuel reaches the incandescent state and is made up of practically pure carbon and ash.

In order to assist in combustion and reduce the amount of smoke to a minimum, it is necessary thoroughly to mix the air supply with the combustible gases, and at the same time maintain the mixture at a high temperature. If the gases come in contact with the cooler plates of the boiler before combustion has been completed, the process will be arrested and smoke will be formed, hence, it is necessary that special provision be made

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for supplying air and mixing it with the gases before they reach the main part of the heating surface of the boiler.

One of the principle obstacles in the way of satisfactory hand firing is its intermittent character. Ordinarily a considerable quantity of coal is fired at one time, which results in the demand for an increased supply of air at a high temperature above the bed of the fuel. Unless this can be supplied until the volatile gases have been burned, and the coal becomes incandescent, smoke will be formed with the usual results. From the above, it is evident that in designing and operating a furnace the following conditions should be observed as far as practicable:

Arrangements should be made for supplying a sufficient volume of air at a temperature as high as possible, above the fire immediately after fresh fuel has been added, and also for mixing it thoroughly with the volatile gases as fast as they are distilled. In addition to this, the mixture of air and gas must be constantly kept at a temperature above the kindling point, and the furnace must be of such proportions that the gases will be completely burned in the time allowed for their passage through it. Some of the practical devices employed for reaching these results are described in a later chapter, and include mechanical stokers, firebrick arches, and various mixing devices depending upon the type of furnace and grade fuel to be burned.

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CHAPTER IV.

BOILER HORSE POWER

The standard established by the committee of judges at the Centennial Exhibition in 1876 as a "boiler horse power" is the evaporation of 30 pounds of water per hour from a feed-water temperature of 100° F. into steam at 70 pounds gauge pressure. This is equivalent to the evaporation of 34.5 pounds from a temperature of 212° F. into steam at atmospheric pressure, and requires 33,305 B. T. U., which for practical purposes we may call 33,000.

The fact that an engine horse power represents work done at the rate of 33,000 foot pounds per minute often leads to more or less confusion among those not familiar with the meaning of the term horse power as applied to the two cases. A boiler of 100 H. P. may supply sufficient steam for a 200 H. P. compound condensing engine of the best make, or it may only be able to provide steam for a 50 or 75 H. P. simple non-condensing engine of old style with plain slide valve. In each case the boiler would be furnishing the same amount of steam and would be running at its normal rating of 100 H. P., but the work delivered by the engine would vary from 50 to 200 H. P., depending upon its water rate. From this it is evident that the term horse power has two different and distinct meanings when applied to engines and boilers. In the first case it means *work* done, and in the second, *pounds of steam* generated.

Equivalent Evaporation.—We have seen from a previous chapter that the "latent heat of evaporation" varies with the pressure, and that the "heat in the liquid" varies both with the pressure, and with the temperature of the feed water. From this it is evident that in making comparisons of the efficiency of boilers working under different conditions, we must first reduce the working conditions of each to a common standard. This standard is called the "equivalent evaporation from and at 212°," and is best explained by the following example:—

A boiler carrying a pressure of 90 pounds gauge, and supplied

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with feed water at a temperature of 60°, evaporates 3,000 pounds of water per hour. What is its equivalent evaporation from and at 212°?

From Table I. we find the temperature of steam at 90 pounds pressure to be 331°, and the "latent heat of evaporation" 881 B. T. U.; then for one pound of steam the "heat in the liquid" above the temperature of the feed water is 331—60=271 B. T. U. and the "total heat" is 271+881=1,152 B. T. U. Therefore, the heat required to raise the temperature of 3,000 pounds of water from 60°, and evaporate it into steam at 90 pounds pressure, is 1,152 + 3,000 = 3,456,000 B. T. U.

Referring again to Table I., we find the "latent heat of evaporation" for steam at atmospheric pressure to be 966 B. T. U.; therefore, 3,456,000÷966=3,577 lb., which is the weight of water that would be evaporated from a temperature of 212° into steam at atmospheric pressure by the same amount of heat. The ratio 3,577 ÷ 3,000 = 1.19 is called the "factor of evaporation."

Table II. gives the factors of evaporation for other conditions of feed-water temperature and steam pressure. To use the table, look in the first column for the temperature of the feed water supplied, then follow to the right until the column corresponding most nearly to the given boiler pressure is reached. The number here found will be the factor of evaporation for the case in hand, and the *actual* quantity of water evaporated, multiplied by the factor thus found will give the equivalent evaporation from and at 212°.

Efficiency.—The general term "boiler efficiency," as commonly used, includes both the boiler and furnace, and is the ratio of the heat contained in the steam, above feed-water temperature, passing through the nozzle in a given time, to the heat contained in the fuel burned during the same period. Briefly stated, it is the ratio of the in-put to out-put, expressed in heat units.

In making an accurate test it is customary to separate the furnace and boiler efficiencies, letting the former represent the ratio of the heat available in the furnace gases to that contained in the fuel burned and the latter, the ratio of the heat absorbed by the water and steam to that available in the furnace gases. It is evident from the above that the efficiency of the furnace depends

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upon the completeness of combustion, and the efficiency of the boiler upon its power of heat absorption.

High efficiency of the boiler plant, as a whole, is of course the result sought in practical work, but a separation of the two is often desirable for purposes of analysis.

The loses in boiler plant operation are due to the incomplete combustion of the furnace gases, the falling of unburned fuel through the grates, the heat carried away with the chimney gases, and radiation from the setting.

Furnace efficiency may be increased by improvements in design and methods of firing, while the boiler efficiency depends largely upon the ratio of heating to grate surface and the character and arrangement of the former.

The combined efficiency of boilers commonly runs from 50 to 70 per cent. A considerable number of tests of a standard water-tube boiler show efficiencies ranging from 67 per cent to 83 per cent.

In the case of small and poorly cared for plants the efficiency will often fall considerably below 50 per cent.

Heating Surface, Grate Area, Etc.

Heating Surface.—The commercial horse power of a well-designed boiler is based upon its heating surface. It has been found by experience that with a properly proportioned grate area and a good chimney draft there is a definite relation between the square feet of heating surface in a boiler and the weight of water which it will evaporate in a given time.

For the best economy, a boiler for power work should be so proportioned as to have at least 1 square foot of heating surface for each 3 pounds of water evaporated from and at 212° per hour. For the smaller sizes of heating boilers the evaporation should not exceed 2 pounds per square foot of heating surface per hour. The first figure gives $34.5 \div 3 = 11.5$ square feet of heating surface per horse power, and the second, $34.5 \div 2 = 17.2$ square feet per horse power.

It is customary for most builders of tubular boilers to rate them on a basis of 12 square feet of heating surface per horse power. This is a safe figure provided the heating surfaces are kept free from soot and ashes, and are so arranged as to produce an even

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flow of the hot gases over them. The interior of the boiler must also be kept free from sediment and incrustation if the best results are to be obtained. As it is practically impossible to run a boiler under these favorable conditions at all times, especially in heating plants of small or medium size, it seems best to make a slight allowance by providing 15 square feet of heating surface

TABLE II.

FACTORS OF EVAPORATION.

Temp're of feed water. Degrees F.	Steam pressure by gauge.																	
	50	60	70	80	90	100	110	120	130	140	150	160	170	180	190	200	210	220
32	1.214	1.216	1.220	1.222	1.225	1.227	1.229	1.231	1.232	1.234	1.236	1.237	1.239	1.240	1.241	1.243	1.244	1.245
40	1.206	1.209	1.212	1.214	1.216	1.219	1.220	1.222	1.224	1.226	1.227	1.229	1.230	1.232	1.233	1.234	1.236	1.237
50	1.195	1.197	1.201	1.204	1.206	1.208	1.210	1.212	1.214	1.215	1.217	1.218	1.220	1.221	1.225	1.224	1.225	1.226
60	1.185	1.188	1.191	1.193	1.196	1.198	1.200	1.202	1.203	1.205	1.207	1.208	1.210	1.211	1.212	1.214	1.215	1.216
70	1.175	1.178	1.180	1.183	1.185	1.187	1.189	1.191	1.193	1.194	1.196	1.197	1.199	1.200	1.202	1.203	1.205	1.206
80	1.164	1.167	1.170	1.173	1.175	1.177	1.179	1.181	1.183	1.184	1.186	1.187	1.189	1.190	1.192	1.193	1.194	1.195
90	1.154	1.157	1.160	1.162	1.165	1.167	1.169	1.170	1.172	1.174	1.176	1.177	1.179	1.180	1.181	1.183	1.184	1.185
100	1.144	1.147	1.150	1.152	1.154	1.156	1.158	1.160	1.162	1.164	1.165	1.167	1.168	1.170	1.171	1.172	1.174	1.175
110	1.133	1.136	1.139	1.142	1.144	1.146	1.148	1.150	1.152	1.153	1.155	1.156	1.158	1.159	1.160	1.162	1.163	1.164
120	1.123	1.126	1.129	1.131	1.133	1.136	1.138	1.140	1.141	1.143	1.145	1.146	1.147	1.149	1.150	1.151	1.153	1.154
130	1.113	1.116	1.118	1.121	1.123	1.125	1.127	1.129	1.130	1.132	1.134	1.136	1.137	1.138	1.140	1.141	1.142	1.144
140	1.102	1.105	1.108	1.110	1.113	1.115	1.117	1.119	1.120	1.122	1.124	1.125	1.127	1.128	1.129	1.131	1.132	1.133
150	1.091	1.095	1.098	1.100	1.102	1.104	1.106	1.108	1.110	1.111	1.113	1.115	1.116	1.118	1.119	1.120	1.121	1.123
160	1.081	1.084	1.087	1.090	1.092	1.094	1.096	1.098	1.100	1.101	1.103	1.104	1.106	1.107	1.108	1.110	1.111	1.112
170	1.070	1.074	1.077	1.079	1.081	1.083	1.085	1.087	1.089	1.091	1.092	1.094	1.095	1.097	1.098	1.099	1.101	1.102
180	1.060	1.063	1.066	1.069	1.071	1.073	1.075	1.077	1.079	1.080	1.082	1.083	1.085	1.086	1.088	1.089	1.090	1.091
190	1.050	1.053	1.056	1.058	1.060	1.063	1.065	1.066	1.068	1.070	1.071	1.073	1.074	1.076	1.077	1.078	1.080	1.081
200	1.039	1.043	1.045	1.048	1.050	1.052	1.054	1.056	1.058	1.059	1.061	1.063	1.064	1.065	1.067	1.068	1.069	1.071
210	1.029	1.032	1.035	1.037	1.040	1.042	1.044	1.046	1.047	1.049	1.051	1.052	1.053	1.055	1.056	1.057	1.059	1.060

per horse power, which has been found safe and to give good results in practice. This also allows a margin for forcing in case of an emergency. With clean tubes and careful firing, a boiler having these proportions may be made to develop a capacity at least 30 per cent above its nominal rating.

All computations and tables relating to tubular boilers which occur in the following pages are based on 15 square feet of heating surface per horse power.

Authorities differ in regard to the proper method of computing the heating surfaces of tubular boilers. In general, all surface is taken which is exposed to the hot gases on one side and to the water on the other. There is some question as to whether the front head should be considered as effective surface because the

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gases in passing from the tubes to the uptake flow away from it instead of against it as they do in the case of the rear head. A safe rule, and one by which tables appearing in a later chapter are computed, is to take one-half the area of the shell, two-thirds of the rear head, less the tube area, and the interior surface of all the tubes.

In the case of water-tube boilers of the best makes, it is customary to allow 10 square feet of heating surface per horse power. This is due to the greater proportion of heating surface exposed directly to the fire, better circulation of water within the boiler, and, as they are largely used for power purposes, they usually receive better care.

Rate of Combustion.—The coal burned per square foot of grate surface per hour in power plants with natural draft will average from 12 to 15 pounds for anthracite, and from 15 to 18 pounds for bituminous. When forced draft is employed, a combustion of 20 to 30 pounds is often obtained with stationary boilers, while a much higher rate is common in locomotive practice. In the case of heating boilers of good size, from 8 to 10 pounds is a fair average; this, however, is often much less in poorly cared for boilers of small size. House heating boilers of the cast-iron sectional type usually burn about 5 pounds of coal per square foot of grate per hour when properly cared for. It is evident that no fixed rule can be given for determining the rate of combustion in any given case, as it depends so largely upon the chimney draft and grade of fuel used.

Rate of Evaporation.—This, in well-proportioned boilers with clean tubes, varies from 8 to 10 pounds of steam per pound of coal; the latter representing more nearly the performance of the best makes of water-tube boilers. In the case of small heating boilers below 20 H. P., the evaporation is more commonly from 6 to 7 pounds per pound of coal. This, like the rate of combustion, is a variable factor, depending upon the character and arrangement of the heating surfaces and upon the proper proportioning of the grate area.

Grate Area.—The required amount of grate area per horse power and the proper ratio of heating surface to grate area vary a good deal, depending upon the character of the fuel and the chimney draft.

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By assuming the probable rates of combustion and evaporation, we may compute the required grate area for any boiler by the equation

$$S = \frac{\text{H. P.} \times 34.5}{E \times C}$$

in which

S = total grate area, in square feet.

E = pounds of water evaporated per pound of coal.

C = pounds of coal burned per square foot of grate per hour.

Table III. gives the approximate grate area per horse power for different rates of evaporation and combustion as computed by the above formula.

TABLE III.

GRATE AREA PER HORSE POWER.

Pounds of steam per pound of coal.	Pounds of coal burned per square foot of grate per hour.				
	8	10	12	15	20
10	.43	.35	.28	.23	.17
9	.48	.38	.32	.25	.19
8	.54	.43	.36	.29	.22
7	.62	.49	.41	.33	.24
6	.72	.58	.48	.38	.29

With an evaporation of 10 pounds of steam per pound of coal, and a combustion of 15 pounds of coal per square foot of grate per hour, which may be taken as the average performance in a first class plant, with water-tube boilers, 0.23 square foot of grate surface would be required per horse power. For a heating plant with tubular boilers, and rates of evaporation and combustion of 8 and 10, respectively, 0.43 square foot of grate surface per horse power would be required.

The difference in efficiency between the boilers in a power plant and those in a heating plant does not depend in any way upon the use made of the steam, but upon the larger size of the boilers, the greater care bestowed upon them, and the employment of more experienced firemen.

Ratio of Heating to Grate Surface.—This in tubular boilers for heating usually varies from 30 to 40 under ordinary conditions. Taking the lower figure and assuming 15 square feet of heating surface per horse power, a grate area of 0.5 square foot

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per horse power is called for, while with the higher figure an area of 0.38 square foot is required. In the case of water-tube and power boilers the ratio is usually somewhat greater, running from 40 to 50 or more.

Taking these figures and assuming 10 square feet of heating surface per horse power, we find the required grate areas to be 0.25 and 0.20 square foot, respectively, all of which correspond very closely with the figures given in Table III. for the corresponding types of boilers. The most efficient ratio in any given case will depend upon the grade of coal to be used, the rate of combustion, and the character and arrangement of the heating surfaces. In any given boiler there is a limit to the quantity of heat which it can absorb economically, so that, if the rate of combustion is increased, the grate area must be correspondingly diminished, else the gases will pass to the chimney at a high temperature and heat will thus be wasted.

Bituminous coal requires a higher ratio between heating and grate surfaces, because the soot deposited upon the tubes reduces their efficiency, and also because the rate of combustion is usually somewhat greater than with anthracite.

Mr. Geo. H. Barrus in his book, "Boiler Tests," states that in a series of three tests on tubular boilers using anthracite coal, he found no gain in efficiency by increasing the ratio over 36 when the rate of combustion was not over 12 pounds of coal per square foot of grate per hour. With bituminous coal he recommends a ratio of 45 or 50 when the rate of combustion is over 10 to 12 pounds of coal per hour.

The ratio of heating to grate surface in sectional cast iron boilers for house heating, ranges from 15 to 25 in the best makes. This low ratio is due to the slow rate of combustion commonly employed.

Coal per Horse Power.—It has been found by experience that a combustion of about 0.3 pound of coal per hour for each square foot of heating surface seems to give the most satisfactory results. This gives an average coal consumption of $15 \times 0.3 = 4.5$ pounds per horse power per hour for tubular boilers and $10 \times 0.3 = 3$ pounds for water-tube boilers. These figures will vary somewhat with the quality of the coal, but will be found useful in making approximations for average boiler practice.

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Computing Boiler Horse Power

Buildings of large size require steam for one or more of the following purposes: Heating; ventilation; hot water; and power for lighting, refrigeration, elevator service and other uses.

Boiler Power for Heating.—The boiler power required for heating may be computed in several different ways. In the case of new buildings it is customary to compute the total heat loss from the building in B. T. U. in the coldest weather, by one of the rules in common use, and divide the results by 33,000. This gives the boiler horse power necessary to evaporate the required amount of steam from and at 212°, but the conditions of temperature and pressure are so similar to this in low-pressure heating that no correction is necessary.

Sometimes it is desired to install a boiler plant in a building where the radiation is already in place. In this case we may use the following relations between steam radiating surface and boiler power, assuming that one boiler horse power will supply.

120 square feet of direct cast iron radiation.
100 square feet of direct wrought iron pipe coils.
60 square feet of indirect cast iron radiation.
20 square feet of steam blast coils.

The boiler power computed in this manner should be increased about 10 per cent to cover the loss by radiation from the steam mains and returns.

Power of Cast Iron Sectional Boilers.—Cast iron sectional, and other heating boilers of similar design, are usually rated by the square feet of direct radiation which they will supply instead of by horse power. The published ratings are usually somewhat in excess of the *economical* capacities of the boilers, and it is better to compute the size required in any given case rather than to rely upon the catalogue rating.

For the best types we may assume an average combustion of 5 to 6 pounds of coal per square foot of grate per hour, and an average efficiency of 60 per cent, which corresponds to about 8,000 B. T. U. per pound of coal available for useful work.

The most efficient rate of combustion, however, will depend to some extent upon the ratio between the heating and grate surfaces. A combustion of about one fourth pound of coal per hour

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for each square foot of heating surface seems to give the best results for boilers of this type, so that by knowing the ratio of heating surface to grate area we may easily compute the most efficient rate of combustion for any given make and size of boiler, and from this determine the necessary grate area.

Example.—The heat loss from a building is 480,000 B. T. U. per hour and we wish to use a sectional boiler in which the ratio of heating surface to grate area is 24. What will be the most efficient rate of combustion, and the required grate area?

Solving, $480{,}000 \div 8{,}000 = 60$ lb. of coal required per hour, and $24 \div 4 = 6$, the most economical rate of combustion to employ. Therefore, the required grate area is $60 \div 6 = 10$ sq. ft.

Boiler Power for Ventilation.—The heat, in thermal units, required to raise any given volume of air through any number of degrees in temperature is equal to

$$\frac{\text{Volume of air in cubic feet} \times \text{Degrees raised}}{55}$$

Knowing the cubic feet of air to be delivered per hour to a building, and the number of degrees its temperature is to be raised, we can easily compute the B. T. U. required, and this divided by 33,000 will give the boiler horse power, as before.

Example.—The heat loss from a building is 1,650,000 B. T. U. per hour. The air supply for ventilation is to be 1,500,000 cubic feet per hour, raised through 70°. What will be the total boiler horse power required?

$$\text{H. P. for heating} = \frac{1{,}650{,}000}{33{,}000} = 50$$

$$\text{H. P. for ventilation} = \frac{1{,}500{,}000 \times 70}{55 \times 33{,}000} = 58$$

$$\text{Total} \qquad 108$$

Boiler Power for Hot Water Heating.—This is easily computed if the quantity of water and its initial and final temperatures are known.

Example.—What boiler horse power will be required to raise the temperature of 500 gallons of water per hour from 50° to 180°?

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$500 \times 8.3 = 4{,}150$ lb. and $180° - 50° = 130°$ rise in temperature, from which it is evident that $4{,}150 \times 130 = 539{,}500$ B. T. U. are required. This calls for $539{,}500 \div 33{,}000 = 16.3$ boiler H. P.

Placing this in the form of an equation we have,

$$\text{H. P.} = \frac{G \times 8.3 \times (T_2 - T_1)}{33{,}000}$$

in which

G = gallons of water to be heated per hour.

T_1 = initial temperature.

T_2 = final temperature.

Boiler Power for Supplying Steam Engines.—Computations for power boilers are based on the steam consumption of the engines. This varies a good deal, even with the same type of engine, depending upon the size and speed, the pressure carried, and the point of cut-off.

Table IV. gives about the average steam consumption per indicated horse power per hour for first class engines of medium size.

TABLE IV.

Type of engine	Pounds of steam per indicated horse power per hour	
	Non-condensing	Condensing
Simple high speed	30 to 34	22 to 26
Simple medium speed	28 " 32	21 " 25
Simple Corliss	26 " 30	20 " 24
Compound high speed	24 " 28	18 " 22
Compound medium speed	23 " 27	17 " 21
Compound Corliss	22 " 26	16 " 20
Compound Corliss of over 500 horse power	20 " 24	14 " 18

The higher figures may be used for engines of 75 to 200 H. P. and the lower for sizes ranging from 200 to 500 H. P. For engines smaller than 75 H. P. they should be increased slightly. The last item is for compound engines of the larger sizes, such as are met with in manufacturing plants.

Having determined the probable weight of steam required per hour for the particular type of engine to be used, and knowing the

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temperature of the feed water, and the boiler pressure to be carried, the weight of steam can be reduced to an equivalent evaporation from and at 212°, and this result divided by 34.5 will give the boiler horse power required.

Example.—What horse power of boiler will be required to supply steam at 100 pounds gauge pressure from feed water at 60° temperature, for a compound-condensing Corliss engine of 200 indicated horse power?

Taking the higher water rate for this type of engine from Table IV., we have $200 \times 20 = 4{,}000$ pounds of steam required per hour. Looking in Table II., we find the factor of evaporation for the conditions given to be 1.19. Therefore, $4{,}000 \times 1.19 = 4{,}760$, which is the equivalent evaporation from and at 212°, and $4{,}760 \div 34.5 = 138$, the boiler horse power required.

If steam is to be used for other purposes, such as the running of feed pumps, etc., the power must be increased accordingly. It is also well to allow a margin for a future increase in the power of the engine, should occasion require it. It computing the power of a boiler in this manner it is well to increase the result from 10 to 20 per cent as a reserve for the reasons given above.

Boiler Power for Pumps.—The boiler power required for running a pump is computed in a similar manner to that already described for an engine. The rating or capacity of a pump, however, is usually expressed in gallons of water per minute raised to a given height, instead of in horse power as in the case of an engine.

The weight of water, in pounds per minute, multiplied by the height in feet, to which it is raised, divided by 33,000 will give the useful or delivered work of the pump expressed in horse power.

The friction of the moving parts of a pump, and of the water flowing through the passages and valves, is so great that under ordinary working conditions not much more than 50 per cent of the indicated horse power of the steam cylinders is utilized in doing useful work. This, together with the fact that steam is not used expansively, calls for a large amount of steam in proportion to the work done as shown by Table V., which gives the average steam consumption of the ordinary duplex pump.

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In measuring the head against which a pump is to operate, take the vertical distance between the surface of the water in the suc-

TABLE V.

AVERAGE STEAM CONSUMPTION OF DUPLEX PUMPS.

Type of pump.	Pounds of steam per hour per delivered horse power.
Simple non-condensing.....	120
Compound non-condensing.....	65
Triple non-condensing.....	40
High duty non-condensing.....	30

tion reservoir and the highest point in the discharge pipe. If it is to deliver against a pressure, as in feeding a boiler, reduce the pressure to "feet head," by dividing the pressure per square inch by 0.4.

Examples.—A pump is to discharge 4,000 pounds of water per minute into a tank 50 feet above it. The surface of the water in the reservoir from which it is taken is 10 feet below the pump. What is the useful work required by the pump?

Solving, $50 + 10 = 60$ ft. = the total head against which the pump is working, and

$$\frac{4{,}000 \times 60}{33{,}000} = 7.2 \text{ H. P.}$$

A boiler feed pump is to deliver 200 pounds of water per minute against a pressure of 150 pounds. The water is taken from a reservoir 5 feet below the pump. What boiler horse power will be required, assuming the pump to use 120 pounds of steam per hour per delivered horse power?

Here $150 \div 0.4 = 375$ ft. head and $375 + 5 = 380$ ft. total head; then,

$$\frac{200 \times 380}{33{,}000} = 2.3 \text{ H. P.}$$

from which the steam required will be $2.3 \times 120 = 276$ lb. per hour.

If the temperature of the feed water is 60°, and the boiler pressure 150 pounds, the factor of evaporation from Table II. will be 1.2, and $276 \times 1.2 = 331$ lb., the equivalent evaporation

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from and at 212°. Therefore, $331 \div 34.5 = 9.6$, the boiler horse power required.

In the above examples it has been assumed that the horizontal runs in the suction and discharge pipes are short and no account has been taken of frictional resistance.

Boiler Power for Electric Lighting.—The power required for electric lighting can be determined approximately by assuming that one horse power of electrical energy will supply the number of lamps of different types as given in Table VI.

TABLE VI.

NUMBER AND TYPE OF LAMPS SUPPLIED BY ONE HORSE POWER.

No. of lamps supplied by one horse power.	Type and power of lamp.
12	16 candle-power incandescent.
6	32 candle-power incandescent.
2.2	half arc, open.
1.5	full arc, open.
1 to 1.5	closed arc.

The efficiency of a medium size generating set (engine and dynamo) including the losses in transmission will be at least 75 per cent, when located in or near the building to be lighted, so that the electric horse power necessary to supply the lamps, divided by 0.75, will give the indicated horse power of the engine. From this the boiler horse power can be determined as already described.

Example.—What boiler horse power will be required to furnish steam for a lighting plant carrying 2,400 incandescent lamps of 16 candle power and 600 of 32 candle power and 40 closed arcs of the larger size? The boiler pressure is to be 100 pounds, and the feed water is to be delivered at an average temperature of 70°. The engines are to be high-speed, compound, non-condensing, using 28 pounds of steam per indicated horse power.

From the above we have,

$$\begin{aligned} 2{,}400 \div 12 &= 200 \\ 600 \div 6 &= 100 \\ 40 \div 1 &= 40 \\ \hline & 340 \end{aligned}$$

This is the horse power of electrical energy required of the

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dynamos, and $340 \div 0.75 = 453$, I. H. P. of engines; hence, $453 \times 28 = 12{,}684$ pounds of steam will be required per hour. The factor of evaporation for 100 pounds steam pressure and 70° temperature of feed water is 1.15; $12{,}684 \times 1.15 = 14{,}586$ pounds, the equivalent evaporation from and at 212°, and

$$14{,}586 \div 34.5 = 422,$$

the boiler horse power required.

The data in Table VI., for incandescent lighting, is based upon the carbon filament lamp. Tungsten lamps, which are rapidly coming into use, require only about one-half the electrical energy for the same lighting capacity.

Calculating Power for Refrigation

The Refrigeration Process.—Mechanical refrigeration is commonly produced by liquifying a gas like ammonia or sulphur-dioxide under a high pressure, and then allowing it to vaporize at a lower pressure.

The process of vaporization absorbs heat the same as when steam is formed in a boiler, and this results in lowering the temperature of the medium surrounding the pipe or chamber in which vaporization takes place.

In the ammonia *compression* machine the gas is first compressed, then passed through cooling coils called the condenser, where it is liquified. From here it is slowly fed into a coil submerged in brine, one end of which is connected with the suction of the compressor. As the small stream of liquid ammonia flows into the coil, which is under a lower pressure, it at once vaporizes, and in so doing absorbs a certain amount of heat from the surrounding brine, thus lowering its temperature. The brine is then circulated through coils arranged along the ceilings and walls of the rooms to be cooled. Sometimes the coils in which vaporization occurs are made to take the place of the brine coils, and the heat is abstracted directly from the air of the room without the use of brine.

In the *absorption* machine, so called, the results are obtained in a somewhat different manner, although the general principles involved are the same.

The capacity of a refrigerating plant is usually expressed in tons of refrigeration or "ice-melting effect." For example, a

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10-ton machine will produce the same cooling effect in 24 hours as the melting of 10 tons of ice, or, in other words, will extract the same amount of heat from the brine as would be required to melt 10 tons of ice into water at a temperature of 32 degrees.

Boiler Power for Refrigeration.—The indicated horse power required per ton of refrigeration depends upon the suction and condenser pressures, which in turn are governed by the temperature and amount of the condensing water used. Under ordinary conditions where condensing water must be obtained at average city prices, the most economical results are obtained with suction pressures ranging from 20 to 30 pounds, and condenser pressures of 140 to 150 pounds gauge.

Under these conditions, 1 indicated horse power at the steam cylinder will produce about 60 pounds of ice-melting effect per hour, or 0.75 ton per 24 hours. This will, of course, vary somewhat with the range of pressure and also with the size and type of machine, but in the absence of more exact data, may be used for approximate results. Another method in common use is to provide 1.5 indicated horse power per ton of refrigeration, which is slightly more than in the previous case. Knowing the indicated horse power of the compressor, the probable steam consumption can be determined from Table IV., and the boiler power computed the same as for a steam engine of the same type and capacity.

If the machine is motor driven, the required power must be increased from 20 to 25 per cent for losses in the generator and motor.

Example.—What boiler horse power wil be required to operate a 50-ton refrigerating plant running 24 hours per day; the compressor being of the steam driven type, with a simple non-condensing engine using 32 pounds of steam per indicated horse power per hour? Temperature of feed water 60 degrees, boiler pressure 80 pounds.

I. H. P. of compressor $= 50 \div 0.75 = 66.6$. Calling it 70, the steam consumption will be $70 \times 32 = 2{,}240$ pounds per hour. The factor of evaporation for the conditions stated is 1.19, from which the boiler horse power is found to be

$$\frac{2{,}240 \times 1.19}{34.5} = 77$$

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If the machine were motor driven the indicated horse power of the engine operating the generator would be $70 \times 1.2 = 84$, and the steam consumption would be figured accordingly.

Power for Ice Making.—If power is required for the actual manufacture of ice, instead of the cooling of brine or other medium, the boiler power as above computed should be doubled. This is made necessary on account of the introduction of other heat losses not present in the simple process of cooling.

Calculating Power for Elevators

Power for Operating Elevators.—The power necessary to operate an elevator depends upon its size, the method of construction and counterbalancing, the speed, and the efficiency. Placing these conditions in the form of an equation, we have,

$$\text{H. P.} = \frac{(W + w) \times S}{E \times 33{,}000}$$

in which

W = weight of *live load.*

w = unbalanced weight of car.

S = speed in feet per minute.

E = efficiency.

In addition to the above, a certain amount of power is required for accelerating the car each time it is started on its upward trip. This, however, is relatively small when compared with the total work done and has been cared for in an approximation as noted below.

The elevators in most common use for passenger service are of the hydraulic and electric types, while for freight work steam and belted elevators may be added; the latter being connected directly with the line shaft in shops and factories.

The general method of computing the power is the same for both hydraulic and electric elevators; but as they differ in detail, it is abvisable to take them up separately.

The live load for a passenger elevator is usually figured on a basis of 60 to 80 pounds per square foot of floor space, and the weight of the elevator itself from 100 to 125 pounds per square foot, which also includes the safety device. These figures will be found ample for cars of ordinary construction, but may be

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exceeded somewhat in the case of metal cars of especially massive design.

Boiler Power for Hydraulic Elevators.—It is common practice with elevators of this type to counterbalance them up to about three fourths of the weight of the car. The speed varies from about 200 to 600 feet per minute, 400 being about the average for office buildings of medium size. The efficiency is about 60 per cent.

In computing the boiler power, it must be taken into consideration that all of the elevators will probably not be running at one time at their maximum capacity, and also that power is required only on the upward trip, as the weight of the car causes it to descend under the control of a suitable braking device.

When there is no definite information at hand, it is customary to compute the power necessary to run all of the elevators at one time under full load, and base the boiler power on 0.7 of this result, which also includes the power required for accelerating the car.

Example.—An office building has three hydraulic elevators, each having a floor space of 30 square feet. What boiler power should be provided, using the average data given below?

Live load, 70 pounds per sq. ft. of floor space; weight of elevator, 100 pounds per sq. ft. of floor space; speed, 400 feet per minute; efficiency, 60 per cent; steam consumption of pumps, 65 pounds per H. P. per hour.

From the above we have,

$W = 30 \times 70 \times 3 = 6{,}300$

$w = 30 \times 100 \times 3 \times 0.25 = 2{,}250.$

Then, for a continuous upward movement with a full load the required horse power would be

$$\frac{(6{,}300 + 2{,}250) \times 400}{0.60 \times 33{,}000} = 172$$

Now, under actual conditions one half of the time is devoted to the downward trips, and the power required is therefore only one half of the above, or 86 H. P. Making allowance for stops at the various floors and for the time that part of the elevators are idle, we may assume that it will be sufficient to provide for 70 per cent of the above, or $0.70 \times 86 = 60$ H. P.

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The steam consumption under the conditions stated would be $60 \times 65 = 3{,}900$ pounds per hour.

Assuming 30 pounds of steam per boiler horse power, which may be taken with sufficient accuracy when the pressure and feed water temperature are not given, we have for the required boiler horse power, $3{,}900 \div 30 = 130$.

For the type of pumps ordinarily used for elevator work the steam consumption given in Table V. may be used with approximate results.

Types of Electric Elevators.—The most common types of electric elevators are the *drum, screw,* and *duplex.* The speeds at which these are commonly run may be taken as 350, 500, and 600 feet per minute, respectively, although for regular work, speeds above 400 feet are not usually found desirable for the average building.

So far as the necessary power is concerned, the drum and duplex types may be considered together. The efficiency of these machines commonly runs from 50 to 70 per cent, although, theoretically, the former is the more efficient type. In practice it is not customary to count on much more than 50 per cent, which gives results on the side of safety.

The method of balancing these elevators differs from the hydraulic, in that the entire weight of the car and from 40 to 50 per cent of the maximum live load is counterbalanced. From this it is evident that with no load the power required to pull the car down is that necessary to raise the counterweight, which is equal to one half the maximum live load, and to overcome the friction of the machine. When the car is half full, it is balanced, and the power required is that due to friction only. At full load the conditions are the same as for an empty car, except the power is required during the upward trip instead of the downward.

Power for Operating Electric Elevators.—From the above it is evident that power may be required for both the upward and downward trips, depending upon the number of people in the car, but it will never be as great at any one time as in the case of the hydraulic elevator.

Example.—Taking the same conditions as in the preceding example, what boiler power will be required to operate electric

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elevators of the drum type, having an efficiency of 50 per cent and a speed of 300 feet per minute?

In this case w, the unbalanced weight of the car, disappears, and the maximum live load is equal to only one half the weight of the people in the car, the other half being counterbalanced, so that $W = 30 \times 70 \times 3 \times 0.5 = 3{,}150$ pounds, from which

$$\text{H. P.} = \frac{3{,}150 \times 300}{0.50 \times 33{,}000} = 57$$

If the full load were carried on both upward and downward trips, or sufficient of it on the downward trip to overbalance the counterweight and the friction of the car, the conditions would be the same as in the case of the hydraulic elevator; that is, power would only be required on the upward trip. This condition, however, does not hold, especially in the case of office buildings, where during the morning hours the maximum loads are on the upward trips with empty, or nearly empty, cars coming down. Under these conditions the power is practically the same on both trips, owing to the necessity of raising the counterweight when the car is descending. This makes it necessary to treat the problem the same as though the machine were raising a continuous load.

Assuming as before that a certain amount of time is required for passengers to enter and leave the car, and that all of the cars will not be running at one time, we may take 70 per cent of the above or $57 \times 0.7 = 40$ as the maximum horse power to be delivered continuously by the motor.

Boiler Power for Electric Elevators.—Assuming efficiencies of 80, 90 and 85 per cent for the motor, generator and engine, respectively, the required indicated horse power of the engine will be

$$\frac{40}{0.80 \times 0.90 \times 0.85} = 62$$

The boiler power will of course depend upon the water rate of the engine. Assuming that a simple non-condensing engine is employed, requiring 30 pounds of steam per indicated horse power per hour, the boiler power will be practically the same as that of the engine, that is, 62 H. P.

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The power required to operate a duplex elevator is practically the same except here a higher speed may be allowed.

The method of balancing a screw machine is practically the same as for the hydraulic type. The efficiency of this machine may be taken as about 70 per cent.

The horse power for driving elevators of this type is calculated the same as for the hydraulic, except for the higher efficiency. After the power of the motor has been computed, the boiler power may be determined as in the preceding example.

Freight elevators are computed in the same way, except they are run at lower speeds, and are built especially to carry the desired load in each particular case. When applying these methods of computation, the engineer should obtain all the data possible regarding the type of machine to be used, the probable speed, efficiency, etc., before proceeding in any particular case; but if any of these data are lacking, the average figures already given may be used with approximate results.

General Considerations

Total Boiler Power.—When a building contains a power plant, the exhaust steam not used for feed-water heating is commonly turned into the heating system. In this case the boiler power for supplying the engines and pumps is first computed, and about 80 per cent of the steam furnished for this purpose may be considered as available in the exhaust for heating purposes. If this amount is less than is required for heating, in the coldest weather, additional boiler power must be provided to make up the deficiency.

Number of Units.—In designing a plant of any considerable size it is better to use two or more smaller boilers rather than a single large one, and it is also well to provide for a certain amount of reserve power for use in case of a breakdown or when it is desired to shut down a boiler for cleaning or inspection. In a heating plant, if sufficient power is provided for the coldest weather, there will be much of the time when only a part of the boilers will be in use, but in the case of a power plant where the demand for steam is practically constant, one or more extra boilers, depending upon the size of the plant, should be kept in reserve.

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CHAPTER V.

TYPES OF BOILERS

Steam boilers are divided into two general classes, known as externally fired and internally fired boilers. These, owing to the course taken by the gases, are again divided into shell or fire-tube boilers, and water-tube boilers.

In the former the heat is applied to the interior of the tubes and in the latter to the exterior.

Externally fired boilers are more commonly used in this country for power and heating, while internally fire boilers are extensively used in marine practice.

There is a great variety of boilers upon the market, but space can only be taken for a few representative types.

Relative Merits.—No particular rule can be laid down for the selection of a boiler. The style adopted must depend largely upon the taste of the purchaser, the maximum price to be paid, the avaliable space, the pressure to be carried, the quality of the feed-water and other conditions under which the boiler is to operate. Experience has shown that when boilers of any of the well-known types are equally well designed and proportioned for the work to be done, and are operated with the same skill, and provided with the same quality of fuel, one type will give about the same economy as another.

The advantages of the fire tube boiler are: first, a greater water capacity, which gives a less rapid fluctuation of the water-line and thus requires less careful attention; second, lower cost of installation; and third, less height than some types of water-tube boilers, which is often of much importance in low basements.

The water-tube boiler has the advantage of greater safety, as the water is divided into small masses, which prevents serious results in the case of rupture. The large amount of heating surface next to the fire increases the transmission of heat and prevents over-heating. The draft area, which is limited in fire-tube boilers, is increased in this type of boiler, thus giving a slower movement to the gases and a greater opportunity for their heat to be absorbed before entering the chimney.

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Another advantage is that riveted joints are removed from the fire. Also, as the steam enters one end of the drum and leaves it at the other the water and steam are more thoroughly separated, and the circulation of water being rapid and in one direction, gives less opportunity for sediment to be deposited on the heating surfaces than in the case of fire-tube boilers. Steam may be raised rapidly, as the water is divided into small streams which pass through the hottest portions of the fire. The loss of heat from the accumulation of ashes and soot is less than in the case of fire-tube boilers, as they do not adhere so readily to the outside of the tubes as to the inside.

Fire-tube Boilers

Horizontal Tubular Boiler.—This boiler consists of a cylindrical shell of wrought iron or steel with fire tubes terminating in the two flat ends. The tubes are usually from two and one half inches to four inches in diameter, depending upon the length of

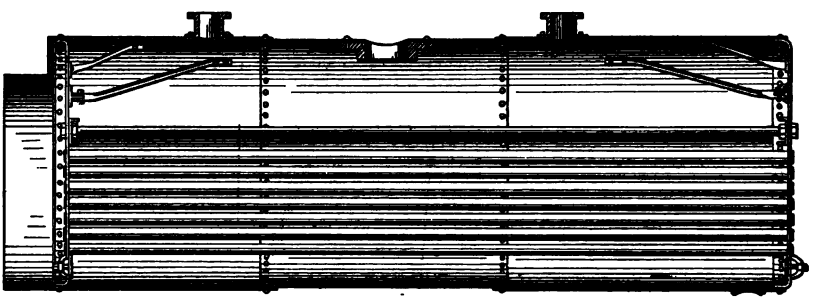

Fig. 1. Horizontal Tubular Boiler.

the shell. The steam space occupies about one third of the volume of the boiler and the remainder is filled with water; the water line should be from four to six inches above the tops of the tubes. The heads below the water line are supported by the tubes, and the remaining portions are stayed by through braces or diagonal stays. Fig. 1 shows a longitudinal section through a boiler of this kind. Horizontal tubular boilers require a brick setting, the proportions and form of which are taken up in a following chapter.

As the horse power of a boiler depends upon its effective heat-

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TABLE VII.

DIMENSIONS OF TUBULAR BOILERS.

Diameter of boiler	Number of tubes	Diameter of tubes	Area through tubes in square feet	Square feet heating surface in heads	Square feet heating surface in one linear foot shell and tubes	Horse power in one linear foot shell and tubes	Length of tubes	Horse power	Rate of evaporation 9 lb. steam per lb. of coal: Size of grate	Size of uptake	Area smoke-pipe, sq. inches	Coal per sq. foot of grate per hour	Rate of evaporation 8 lb. steam per lb. of coal: Size of grate	Size of uptake	Area smoke-pipe, sq. inches	Coal per sq. foot of grate per hour
30	28	2½	0.78	2.8	20.7	1.37	6	8.5	.. ..	.....	....	.....	24 by 36	10 by 14	140	6
							7	9.9	.....	.....	....	.....	24 " 36	10 " 14	140	7
							8	11.2	.....	.....	....	.....	24 " 36	10 " 14	140	8
							9	12.6	.....	.....	....	.....	24 " 42	10 " 14	140	8
							10	14.0	Use fig	ures in	last	four	24 " 42	10 " 14	140	8½
36	34	2½	0.97	4.0	25.0	1.67	8	13.6	column	s for 80	inc	h and	30 " 36	10 " 16	160	8½
							9	15.3	36-inch	boilers			30 " 42	10 " 18	180	8½
							10	16.9	.....	.....	....	.....	30 " 42	10 " 18	180	9
							11	18.6	.....	.....	....	.....	30 " 48	10 " 20	200	9
							12	20.9	.....	.....	....	.....	30 " 48	10 " 20	200	9½
42	34	3	1.44	5.3	30.2	2.0	9	18.5	36 by 36	10 by 20	180	9	36 " 42	10 " 20	200	8½
							10	20.5	36 " 36	10 " 20	180	9½	36 " 42	10 " 20	200	9
							11	22.5	36 " 36	10 " 20	180	10½	36 " 48	10 " 25	250	9
							12	24.5	36 " 36	10 " 20	180	11½	36 " 48	10 " 25	250	9½
							13	26.5	36 " 42	10 " 22	200	11	36 " 48	10 " 25	250	10½
							14	28.5	36 " 42	10 " 22	200	12	36 " 54	10 " 28	280	10½
48	44	3	1.86	7.2	38.4	2.56	10	30.4	42 " 36	10 " 22	200	11	42 " 48	10 " 28	280	9
							11	33.2	42 " 36	10 " 22	200	12	42 " 48	10 " 28	280	10
							12	35.7	42 " 42	10 " 25	220	11½	42 " 54	10 " 32	320	10
							13	38.3	42 " 42	10 " 25	220	12	42 " 54	10 " 32	320	10½
							14	40.8	42 " 42	10 " 25	220	12½	42 " 60	10 " 36	360	10½
							15	43.4	42 " 48	10 " 28	250	12	42 " 60	10 " 36	360	11
							16	45.9	42 " 48	10 " 28	250	12½	42 " 60	10 " 36	360	11
54	54	3	2.28	9.3	46.4	3.10	11	34.6	48 " 42	10 " 28	250	9½	48 " 54	10 " 38	380	8½
							12	37.7	48 " 42	10 " 28	250	10	48 " 54	10 " 38	380	9
							13	40.8	48 " 42	10 " 28	250	11	48 " 54	10 " 38	380	10
							14	43.9	48 " 42	10 " 28	250	12	48 " 54	10 " 38	380	10½
							15	47.0	48 " 48	10 " 33	290	11½	48 " 60	10 " 40	400	10
							16	50.1	48 " 54	10 " 38	320	12	48 " 60	10 " 40	400	11
	46	3½	2.67	8.8	46.3	3.10	17	53.0	48 " 54	10 " 38	320	12½	48 " 60	10 " 40	400	11½
60	72	3	3.04	11.2	59.6	4.0	12	48.4	54 " 48	10 " 38	320	10	54 " 60	12 " 40	460	9½
							13	52.4	54 " 48	10 " 38	320	10½	54 " 60	12 " 40	460	10
							14	56.4	54 " 48	10 " 38	320	12	54 " 60	12 " 40	460	11
							15	60.4	54 " 54	12 " 35	370	12	54 " 66	12 " 42	500	10½
							16	64.4	54 " 54	12 " 35	370	12½	54 " 66	12 " 42	500	11½
	64	3½	3.71	10.5	62.4	4.16	17	71.4	54 " 60	12 " 40	400	12½	54 " 72	12 " 48	550	11½
							18	75.6	54 " 60	12 " 40	400	13½	54 " 72	12 " 48	550	12
66	90	3	3.80	13.5	74.2	4.95	14	70.1	60 " 54	12 " 40	400	12	60 " 66	12 " 48	500	11
							15	75.0	60 " 60	12 " 44	450	12	60 " 72	12 " 52	620	11
							16	80.0	60 " 60	12 " 44	450	12½	60 " 72	12 " 52	620	12
	78	3½	4.52	12.6	75.2	5.10	17	86.0	60 " 60	12 " 44	450	13	66 " 72	12 " 56	670	11½
							18	91.1	60 " 60	12 " 44	450	13½	66 " 72	12 " 56	670	12
							19	96.2	60 " 66	12 " 48	500	13	66 " 72	12 " 56	670	12½
	62	4	4.32	12.4	69.2	4.60	20	93.1	60 " 66	12 " 48	500	13	66 " 72	12 " 56	670	12
72	114	3	4.81	15.7	92.5	6.16	14	87.4	66 " 60	12 " 48	500	12	66 " 72	12 " 56	670	11½
							15	93.6	66 " 60	12 " 48	500	13	66 " 72	12 " 56	670	12
							16	99.7	66 " 60	12 " 48	500	14	72 " 72	12 " 62	740	12
	98	3½	5.75	15.0	93.0	6.20	17	106.4	66 " 66	12 " 52	540	13½	72 " 72	12 " 62	740	13
							18	112.6	66 " 66	12 " 52	540	14	72 " 72	12 " 62	790	13
							19	118.8	66 " 66	12 " 52	540	14½	72 " 72	12 " 62	790	14
	72	4	5.02	15.0	79.7	5.31	20	107.3	66 " 60	12 " 48	500	14½	72 " 72	12 " 62	740	13

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ing surface, it follows that with any given diameter and length of shell the power will vary with the size and number of tubes. If the tubes are placed too close together and so near the shell as to

TABLE VIII.

DIMENSIONS OF LIGHT SETTINGS FOR BOILERS USED IN HEATING.

Dia. of boiler.	Add the figures given below to length of tubes, to find length of brick setting.		Width of setting, 1 boiler.		Width of setting, 2 boilers.		Width of setting, 3 boilers.		Width of setting, 4 boilers.	
In.	Ft.	In.	Ft.	In.	Ft.	In.	Ft.	In.	Ft.	In.
30	3	2	5	8	9	8	13	8	17	8
36	3	4	6	2	10	10	15	6	20	2
42	3	4	6	8	11	10	17	0	22	2
48	3	10	7	2	12	10	18	6	24	2
54	3	10	7	8	14	0	20	4	26	8
60	3	10	8	2	15	0	21	10	28	8
66	3	10	9	10	17	8	25	6	33	4
			9	4	16	8	24	0	31	4
72	3	10	* 10	4	18	8	26	6	34	10
			9	10	17	8	25	0	32	10

* Upper figures to be used when width of grate equals diameter of boiler; lower figures when it is 6 inches less.

interfere with the free circulation of the water, the efficiency of their heating surface will be impaired and the performance of the boiler will fall below its rating.

The number and size of tubes given in Table VII. for boilers of

TABLE IX.

DIMENSIONS OF HEAVY SETTINGS FOR POWER BOILERS.

Dia. of boiler	Add the figures given below to length of tubes, to find length of brick setting.		Width of setting, 1 boiler.		Width of setting, 2 boilers.		Width of setting, 3 boilers.		Width of setting, 4 boilers.	
In.	Ft.	In.	Ft.	In.	Ft.	In.	Ft.	In.	Ft.	In.
30	3	6	5	8	9	8	13	8	17	8
36	3	8	6	10	11	8	16	6	22	4
42	3	8	7	4	12	8	18	0	23	4
48	4	2	7	10	13	8	19	6	25	4
54	4	2	9	0	15	8	22	4	29	0
60	4	2	9	6	16	8	23	10	31	0
66	4	2	10	0	17	8	25	4	33	0
72	4	2	10	6	18	8	26	10	35	0

different diameters have been found by experience to give satisfactory results, and may be used with safety. The sizes of grates given for different rates of combustion and evaporation will be found convenient for reference. The sizes of uptake and smoke pipe in each case are proportioned for the corresponding size of grate and rate of combustion.

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Tables VIII. and IX. give the over-all dimensions for boilers of different sizes with light and heavy settings, as called for in Table XXVII.

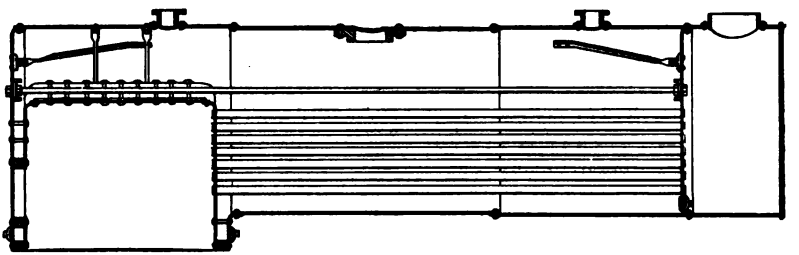

Fig. 2. Locomotive Type Boiler.

Locomotive Boiler.—Another form of tubular boiler in quite common use for stationary purposes is the "locomotive" type. This is an internally fired boiler, and consists of a cylindrical shell, called the barrel, attached to a fire box as in Fig. 2. The fire box proper is usually from three to four inches smaller

TABLE X.

DIMENSIONS OF LOCOMOTIVE TYPE BOILERS.

Nominal horse power.....	18	25	40	50	75
Diameter of waist.....	36 in.	42 in.	48 in.	54 in.	60 in.
Length over all.....	13 ft.	14 ft. 4 in.	17 ft. 6 in.	18 ft. 6 in.	21 ft.
Height of boiler.....	4 ft.	4 ft. 8 in.	5 ft. 4 in.	5 ft. 10 in.	6 ft. 6 in.
Number of tubes, 3 in. outside diameter.....	28	38	49	60	80
Length of tubes.....	7 ft.	8 ft.	10 ft. 6 in.	11 ft.	13 ft.
Area through tubes.....	1.23 sq. ft.	1.65 sq. ft.	2.27 sq. ft.	2.47 sq. ft.	3.38 sq. ft.
Length of furnace.....	48 in.	50 in.	54 in.	60 in.	60 in.
Width of furnace.....	30 in.	36 in.	42 in.	48 in.	54 in.

than the outer shell, being secured to the latter by means of stay-bolts. The spaces between the inner and outer shells are called the water-legs. The tubes are expanded into the front of the fire box and into the rear head of the barrel. Table X. gives the principal dimensions for standard locomotive boilers.

This type of boiler requires a brick ash-pit, the walls of which act as a support for the furnace. The other end of the boiler rests in a cast iron saddle, supported upon either a brick pier or an

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iron column. The shell and furnace are usually covered with a lagging of asbestos or magnesia.

Vertical Fire-Tube Boiler.— A boiler of this type is shown in Fig. 3. Since it is internally fired it requires no setting except a foundation, and is frequently used where floor space is limited and the power required is small. These boilers are commonly built in sizes

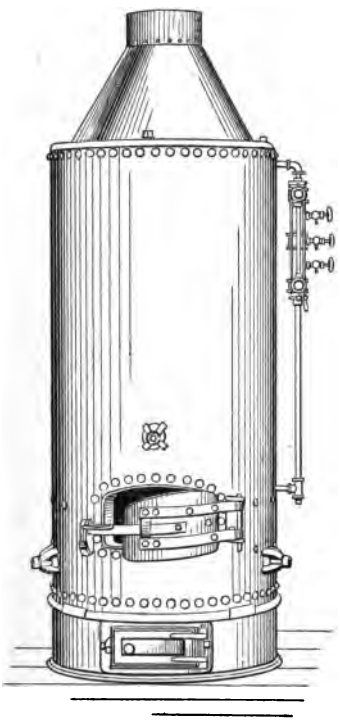

Fig. 3. Vertical Fire-Tube. Boiler.

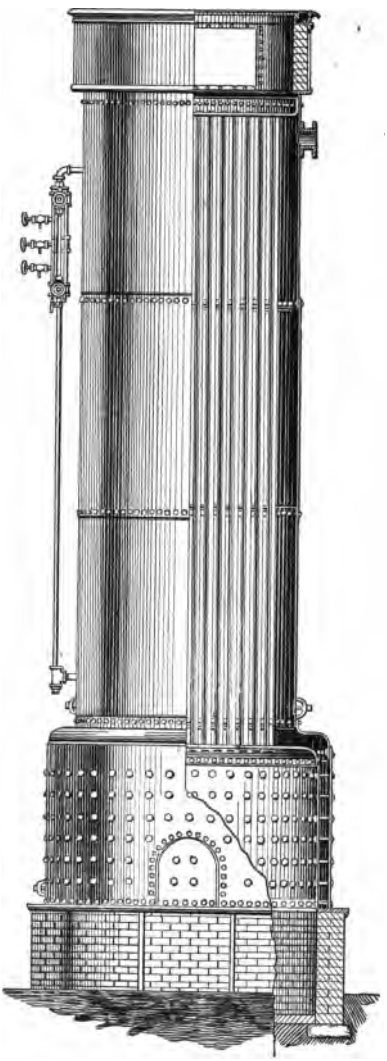

Fig. 4. Manning Boiler.

ranging from five to twenty-five horse power. Table XI. gives the principal dimensions.

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Manning Boiler.—A sectional view of the Manning boiler is shown in Fig. 4. This is similar to the one just described, but is built in larger sizes. The general construction of the boiler is shown in the cut. The special features comprise the means for taking care of the expansion of the tubes and shell. This is provided for by means of a double flanged head connecting the barrel or cylindrical part of the boiler with the outside of the

TABLE XI.

DIMENSIONS OF VERTICAL FIRE-TUBE BOILERS.

Rating—15 square feet of heating surface per horse power.

Diameter of shell.	Diameter of furnace.	No. of tubes 2 inches outside diam.	Area of tube opening.	Length of tubes.	Nominal horse power.	Height over all, including base and bonnet.	Diameter of smoke-pipe.
In.	In.		Sq. ft.	Ft. In.		Ft. In.	In.
30	24	45	.75	3 0	5.8	8	
30	24	45	.75	3 6	6.5	8 6	
30	24	45	.75	4 0	7.3	9	12
30	24	45	.75	4 6	8.	9 6	
30	24	45	.75	5 0	8.7	10	
36	30	65	1.09	4 0	10.3	9	
36	30	65	1.09	4 6	11.4	9 6	
36	30	65	1.09	5 0	12.5	10	14
36	30	65	1.09	5 6	13.5	10 6	
36	30	65	1.09	6 0	14.6	11	
42	36	100	1.67	4 6	17.	9 8	
42	36	100	1.67	5 0	18.7	10 2	
42	36	100	1.67	5 6	20.3	10 8	16
42	36	100	1.67	6 0	21.9	11 2	
48	42	120	2.	5 0	22.5	10 2	
48	42	120	2.	5 6	24.5	10 8	18
48	42	120	2.	6 0	26.43	11 2	

fire box. The double flange forms an expansion joint and also allows a larger diameter of fire box which gives the desired ratio of grate to heating surface. The crown sheet is well above the grate, thus giving ample room for the complete combustion of the gases within the furnace. Special handholes are provided in the different parts to allow for the proper cleaning and inspection of the boiler. The products of combustion enter a smoke box at the top, from which they are conveyed to the chimney through a pipe which may be attached either at the top or side of the box. The water-level is so far below the upper tube sheet that there is

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ample steam space, and the highly heated tubes passing through it enable the boiler to furnish dry steam under practically all conditions of service.

Table XII. gives dimensions of the Manning boiler.

TABLE XII.

DIMENSIONS OF MANNING BOILERS.

Diameter of shell.	Diameter of furnace, outside.	Diameter of furnace, inside.	Number of tubes.	Length of tubes.	Diameter of tubes, outside.	Horse power.	Total height of boiler from floor line to top of bonnet.
In.	In.	In.		Ft.	In.		Ft. In.
38	55	48	90	11	2	36.8	19 4½
44	61	54	86	13	2½	52.1	21 4½
48	65	58	112	15	"	76.4	23 6½
50	67	60	124	15	"	84.2	23 8½
56	73	66	152	15	"	102.7	23 10½
61	79	72	184	15	"	123.9	24 1½
72	98	90	260	15	"	173.4	24 5

Water-tube Boilers

Aultman & Tailor Boiler (Babcock & Wilcox Type.)—This boiler consists of a group of wrought-iron tubes placed in an inclined position as shown, with a horizontal steam and water drum above them. (Fig. 5.)

The tubes are connected with the drum by means of headers and pipes, nearly vertical, at each end. The drum is made up of heavy iron or steel plates, and is usually of the form shown. There are several patterns made for different classes of work; among them the "cross-drum," in which the drum is placed at the rear in a transverse position and connected directly with the rear row of vertical tubes. This pattern requires less head room and is especially adapted to low basements. The boiler is suspended from iron girders which rest on iron columns and is independent of the brickwork. The fire is under the front end of the inclined tubes and the gases are guided by baffle-plates and bridge wall up between the tubes into the combustion chamber, which is under the drum; thence down through another section of tubes and

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then up again a second time, and to the chimney through an opening in the rear wall.

The tubes in the different rows are "staggered" so that the gases are broken up in passing among them and a large proportion of their heat is absorbed. As the water in the tubes becomes heated, it rises toward the higher end, where it becomes partly converted into steam, and rises into the drum where the water

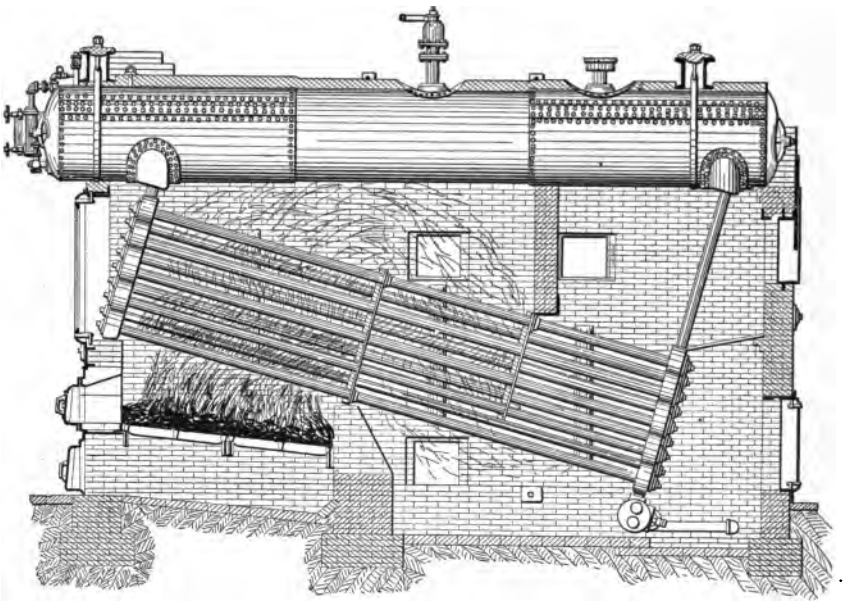

Fig. 5. Aultman and Taylor Boiler. (Babcock and Wilcox Type.)

and steam are separated. The water flows through the drum and down the tubes at the rear, thus giving a circulation which is rapid and efficient. A mud drum of cast iron is connected with the tubes at their lowest point, forming a receptacle for sediment, from which it can be blown out from time to time as required. The steam is taken out at the top of the steam drum near the back end of the boiler, and to insure its dryness a perforated dry-pipe is connected with the nozzle inside the drum. The feedwater is discharged into the steam and water drum near the bottom and under a hood or deflector which tends to mix it with the hotter water flowing toward the rear of the boiler.

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The Caldwell, Gill, National, and Standard are names of boilers similar in principle to the Aultman and Taylor, varying chiefly in details of construction. The design of these boilers is such that the relative proportions may be varied to meet the requirements of any particular case.

The length of tubes usually varies from 10 to 18 feet; they are placed in vertical and horizontal rows, six inches on centers vertically and seven inches horizontally. Single boilers vary from five to 14 tubes in width and from seven to 14 or more in height. The tubes having an outside diameter of four inches, present a heating surface of one square foot for each foot in length.

Table XIII. gives the horse power and approximate over-all dimensions for a number of the more common sizes. This table applies to the regular pattern with *inclined* headers. The height depends upon the number of tubes in the vertical rows. For other sizes than those given in the table, add or subtract six inches for each vertical row, as the case may be.

For *vertical* header boilers, subtract two feet six inches from the lengths given in the table.

For *cross drum* boilers, use the lengths in the table, and for the heights subtract four feet from the tabulated heights. When more than 14 sections in width are required, two boilers should be used, set in a battery. The width of the battery will be twice that of a single boiler, less 17 inches. Spaces of at least two feet should be left at one side and at the rear of single boilers, and at both sides of double boilers, for reaching the cleanout doors, and space should be left in front for drawing the tubes. The smoke connections are taken from the rear of the setting.

Heine Boiler.—Fig. 6 shows a section of the Heine boiler which is similar in many respects to the one just described. It is a water-tube, but not a sectional boiler. The tubes are of wrought iron, expanded into the inside faces of the water legs at either end. The parallel plates of the water legs are supported by stay-bolts passing through them, and are joined at the top to the shell of the drum by flanged and riveted joints. The heads of the drum are spherical and require no staying. A hand-hole is placed over the end of each tube in the head plate, giving access to the tube and a chance for removal. The products of

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combustion pass over the bridge wall into the combustion chamber, then back toward the front between two rows of tiling which are supported upon the upper and lower rows of tubes,

TABLE XIII.

DIMENSIONS OF AULTMAN AND TAYLOR BOILERS WITH INCLINED HEADERS.

Rating — 10 square feet of heating surface per horse power.

Length of tubes,	Number of tubes high	Horse power per vertical section	Number of sections wide	Total horse power	Length of setting,	Width of setting	Height to top of steam outlet
Ft.					Ft.	Ft. In.	Ft. In.
12	7	8.4	6	50.4	17	6 8	15
			8	67.2		7 10	
			10	84.0		9 0	
			12	100.8		10 2	
			14	117.6		11 4	
	9	10.8	6	64.8		6 8	15 6
			8	86.4		7 10	
			10	108.0		9 0	
			12	129.6		10 2	
			14	151.2		11 4	
14	7	9.8	6	58.8	19	6 8	15
			8	78.4		7 10	
			10	98.0		9 0	
			12	117.6		10 2	
			14	137.2		11 4	
	9	12.6	6	75.6		6 8	15 6
			8	100.8		7 10	
			10	126.0		9 0	
			12	151.2		10 2	
			14	176.4		11 4	
16	7	11.2	8	89.6	21	7 10	15
			10	112.0		9 0	
			12	134.4		10 2	
			14	156.8		11 4	
	9	14.4	8	91.2		7 10	15 6
			10	144.0		9 0	
			12	172.8		10 2	
			14	201.6		11 4	
18	7	12.6	8	100.8	23	7 10	15
			10	126.0		9 0	
			12	151.2		10 2	
			14	176.4		11 4	
	9	16.2	8	129.6		7 10	15 6
			10	162.0		9 0	
			12	194.4		10 2	
			14	226.8		11 4	

then upward through the space between the upper row of tiling and the front water leg, and back underneath the water and steam drum, and are finally taken off at the rear, through an iron bonnet or hood. The boiler is supported on brickwork by the two water legs. The feedwater enters the front head of the drum

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and is discharged into the open end of a cylinder or mud drum, having the same inclination as the tubes. The sediment collects in the lower end of this drum and can be discharged through the blow-off pipe, which is connected at this point. A second blow-off or drain is connected with the rear water leg at its lowest point. The water becoming hot in the mud drum rises and flows in a re-

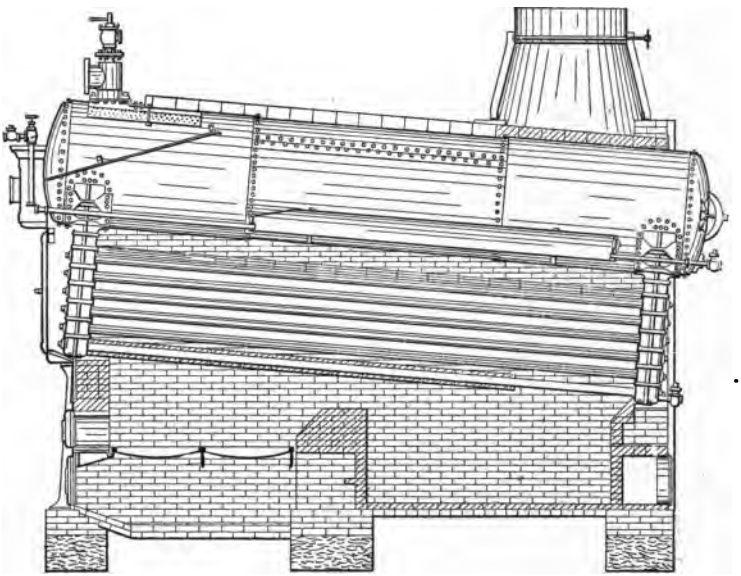

Fig. 6. Heine Boiler.

verse direction toward the open end, then passes in a thin sheet to the rear of the water drum, where it enters the water leg and flows from here through the tubes to the front of the boiler. While in the tubes, it is partially converted into steam, which rises through the water at the front of the drum into the steam space. A deflecting plate is here provided to prevent the water from being carried into the perforated steam pipe above. The Franklin, Keeler, Edge Moor, and Geary boilers have the same general arrangement of tubes and steam drums, while the Zell boiler has features common to both the Aultman and Taylor and the Heine boilers. These boilers range from 69 to 500 horse power each. Any number may be placed side by side in a battery without passageways between them.

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To find the width of a battery of boilers of any given size, multiply the width of furnace given in Table XIV. by the number to be installed and add 20 inches for each outside wall and 27 inches

TABLE XIV.

DIMENSIONS OF HEINE BOILERS.

Rating — 10 square feet of heating surface per horse power.

Horse power	Length of tubes	Width of furnace	Square feet heating surface	Length of setting	Height	Width of setting
	Ft.	In.		Ft. In.	Ft. In.	Ft. In.
69	14	46	688	15 4	12 10	7 2
90	16	53	903	17 4	13 5	7 9
to	to		to	to	to	
126	18		1263	19 4	14 0	
127	16	60	1273	17 4	13 6	8 4
to	to		to	to	to	
171	18		1714	19 4	14 1	
142	16	67	1420	17 4	14 8½	8 11
to	to		to	to	to	
191	18		1918	19 4	15 8½	
156	16	74	1564	17 4	14 2	9 6
to	to		to	to	to	
210	13		2106	19 4	14 9	
171	16	81	1716	17 4	15 10	10 1
to	to		to	to	to	
230	18		2303	19 4	16 10	
225	16	88	2244	17 4	16 8	10 8
to	to		to	to	to	
293	13		2931	19 4	17 3	
242	16	95	2417	17 4	18 3	11 3
to	to		to	to	to	
316	13		3160	19 4	18 10	
281	16	109	2803	17 4	15 7	12 5
to	to		to	to	to	
367	18		3669	19 4	16 2	
298	16	116	2978	17 4	15 8	13 0
to	to		to	to	to	
389	18		3892	19 4	16 3	
372	16	123	3724	17 4	17 1½	13 7
to	to		to	to	to	
478	18		4785	19 4	17 8½	
391	16	130	3912	17 4	17 1½	14 2
to	to		to	to	to	
500	18		5002	19 4	17 8½	

for each division wall. If the space is limited, 20-inch division walls can be used. A passageway of at least two feet should be left at the rear of the setting for reaching the cleanout doors,

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and a space in front for drawing the tubes. The smoke connections are on top near the rear. Table XIV. gives the principal dimensions for a variety of sizes.

Root Boiler.—The Root boiler, Fig. 7, is made up of 4-inch wrought-iron tubes expanded in pairs into cast-iron headers at

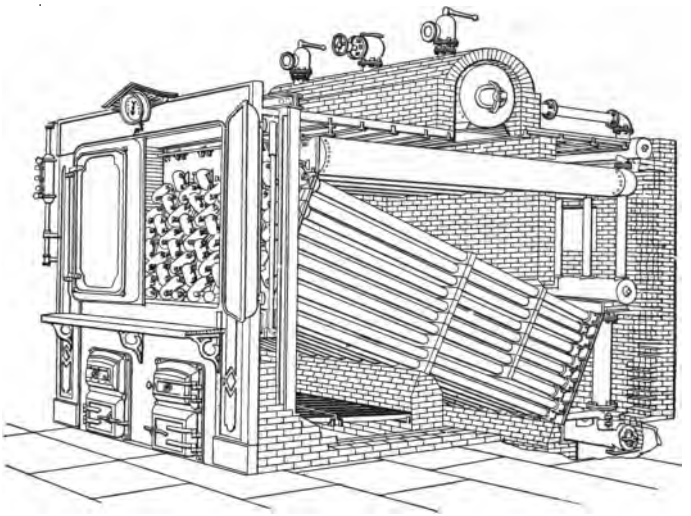

Fig. 7. Root Boiler.

both ends. Each tube is connected to one in the pair above it by a return bend, so that a continuous flow of water is obtained from the bottom to the top of the section. The boiler is made up of several of the vertical sections placed side by side. The tubes in the sections are "staggered," and provision is made for unequal expansion. Each section has its separate overhead drum to receive its discharge of steam and water. At the rear of each overhead drum is a vertical pipe leading to a cross drum beneath, which serves as a common connection for all of the sections. The feed water enters this drum and mixes with the hot water from above. The cross drum is also connected by vertical pipes with another drum below and parallel to it, which serves as a mud drum. The feed water first falls into the mud drum, then rises through the tubes to the overhead drums, where the steam separates and finds its way to the large steam drum which is

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shown across the top of the boiler. Dry steam is taken from the steam drum, the water line being at about the middle of the steam and water drums below. The gases pass among the tubes three times before reaching the chimney. A steam jet is used for cleaning the tubes, and numerous openings with sliding doors are provided for this purpose as well as to give access for inspection and repairs.

These boilers are built in sizes up to 400 horse power, ranging

TABLE XV.

DIMENSIONS OF ROOT BOILERS, CROSS-DRUM PATTERN.

Rating—10 square feet of heating surface per horse power

Number of tubes wide.	Number of tubes high.	Horse power.	Length of setting.	Width of setting.	Approximate height to top of steam outlet.
Fifteen-foot tubes.					
			Ft. In.	Ft. In.	Ft. In.
4	6	36	19 2	5 2	13 0
6	7	63	19 4	6 6	13 6
8	8	96	19 7	7 10	14 0
10	9	135	19 9	9 2	15 0
12	10	180	19 11	10 6	15 6
14	12	252	20 4	11 10	17 0
Eighteen-foot tubes.					
6	7	76	22 2	7 2	14 6
8	8	115	22 5	8 6	15 0
10	9	162	22 7	9 10	16 0
12	10	216	22 9	11 2	16 6
14	12	302	23 2	12 6	18 0
16	13	375	23 4	13 10	18 6

from four to 16 tubes in width and from six to 13 in height. Each row of tubes adds approximately six and one-half inches to the height and eight inches to the width. The tubes are four inches in diameter and the lengths commonly used run from 15 to 18 feet. Space should be left at the rear and one side of a single boiler for reaching the cleanout doors. When two boilers are set in a battery a passageway should be left at each side. Space should be provided in front for drawing the tubes. Table XV. gives the over-all dimensions for a few standard sizes of the cross-drum pattern shown in Fig. 7.

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The width of two of these boilers set in a battery is equal to twice the width of a single boiler, less 9 inches for 15-foot tubes, and less 17 inches for 18-foot tubes as given in the above table. Various combinations of width, length and height other than those given can be made up to meet special conditions.

The smoke connections are taken from the rear.

Robb-Mumford Boiler.—The Robb-Mumford boiler, shown

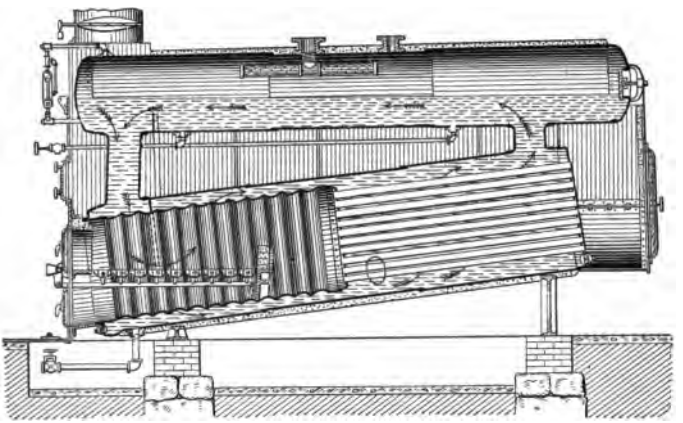

Fig. 8. Robb-Mumford Boiler.

in Fig. 8, is an internally fired water-tube boiler. It consists of two cylindrical shells, the lower one containing a round furnace and tubes, and the upper one forming the steam drum, the two being connected by two necks. The lower shell has an incline of about 1 inch per foot from the horizontal, for the purpose of promoting circulation and draught, and also for convenience in washing out the lower shell. Combustion takes place in the furnace, which is surrounded entirely by water, and the gases after leaving the furnace, pass through the tubes and return between the lower and upper shells, (this space being enclosed by a steel casing), to the outlet at the front of the boiler. The water, together with the steam formed by contact with the furnace and tubes, circulates rapidly up the rear neck into the steam drum, where the steam is released, the water passing along the upper drum towards the front of the boiler and down the front neck; a semi-circular baffle plate around the furnace causing the down-

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flowing water to circulate to the lowest part of the lower shell under the furnace.

The outer casing which incloses the space between the lower and upper shells, including the rear smoke box and smoke outlet, as well as the front and back, is constructed of steel plate, with angle iron stiffeners, the various sections being bolted together for convenient removal. The inside of the steel case including the rear smoke chamber is lined with asbestos air cell blocks fitted in between the angle iron stiffeners. The top of the upper drum and bottom of the lower shell are also covered with non-conducting material after the boiler is erected.

As every part of the boiler is cylindrical except the tube sheets, no staying is required. The furnace is of the corrugated type for high pressures, or with Adamson expansion joints for lower pressures, providing perfectly for unequal expansion and contraction, and requiring no staying. The water in the boiler is comparatively small in volume, and well subdivided, so that a rupture is no more dangerous than in a sectional boiler.

Table XVI. gives the over-all dimensions of the standard sizes

TABLE XVI.

DIMENSIONS OF ROBB-MUMFORD BOILERS.

Rating — 10½ square feet of heating surface per horse power.

Horse power.	Total width.	Total length.	Height to Steam outlet.
	Ft.	Ft. In.	Ft. In.
50	54	14 6	9 6
65	54	18 6	9 6
80	60	18 6	10 10
100	60	21 0	10 10
125	66	21 0	11 8
150	72	21 0	12 8
200	79	23 0	13 10

of this type of boiler. The boilers, when set in battery, are placed side by side, so that the width in any case may be found by simply multiplying the width of a single boiler by the number used. The large open spaces underneath and between the lower and upper drums give access to the handholes and all the interior parts of the boiler. A space of about 3 feet should be left at the rear for reaching the cleanout doors.

Stirling Boiler.—The Stirling boiler, Fig. 9, consists of three steam and water drums at the top and a large mud drum at the

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bottom. All of the upper drums are connected by steam circulating tubes, and the front and middle drums are connected by water circulating tubes also. The feed water enters the rear upper drum and steam is taken off from the middle drum. The lower drum acts as a receptacle for sediment and is connected

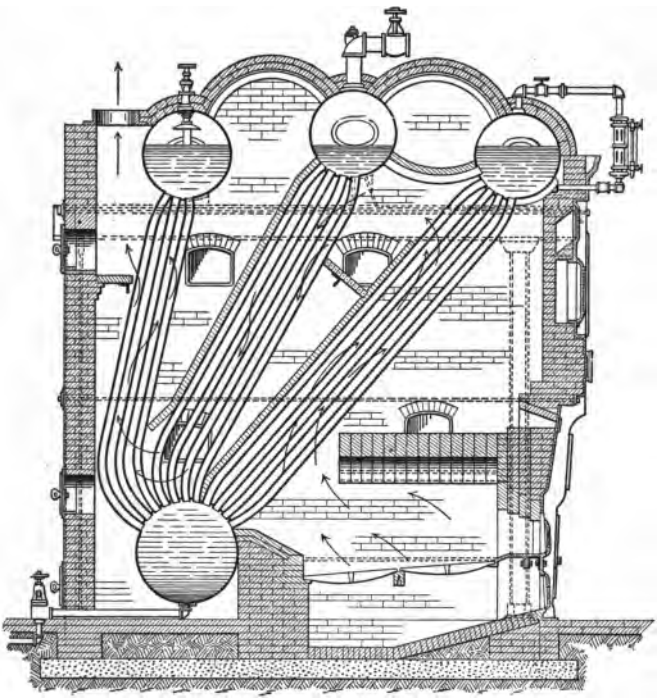

Fig. 9. Stirling Boiler.

with the blow-off pipe. Each set of tubes is separated from the others by fire-brick tile, so arranged that the gases rise among the first set, are then guided downward through the second set and then rise among the third set, and are taken off at the rear. The feed water as stated, enters the rear upper drum, then passes down the rear tubes to the mud drum, and during its passage is heated to within a few degrees of that in the front tubes and moves in a direction opposite to that of the furnace gases, which pass into the stack at the point where the coolest water enters the boiler.

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These boilers are made up in a great variety of forms to meet almost any requirement as regards the available space. Table XVII. gives some of the more common sizes. There are, how-

TABLE XVII.

DIMENSIONS OF STIRLING BOILERS.

Rating — 10 square feet of heating surface per horse power.

Horse power	Class "A" Width		Horse power	Class "B" Width		Horse power	Class "B" (low) Width	
	Single	Battery		Single	Battery		Single	Battery
115	7 ft.	13 ft.	47	5 ft. 6 in.	10 ft.	47	5 ft. 6 in.	10 ft.
143	8 "	15 "	57	6 "	11 "	57	6 "	11 "
172	9 "	17 "	66	6 " 6 "	12 "	66	6 " 6 "	12 "
201	10 "	19 "	75	7 "	13 "	75	7 "	13 "
230	11 "	21 "	86	7 " 6 "	14 "	86	7 " 6 "	14 "
258	12 "	23 "	115	9 "	17 "	115	9 "	17 "
287	13 "	25 "	143	10 " 6 "	20 "	143	10 " 6 "	20 "
316	14 "	27 "	172	12 "	23 "	172	12 "	23 "
345	15 "	29 "	198	13 " 6 "	26 "	198	13 " 6 "	26 "
373	16 "	31 "	226	15 "	29 "	226	15 "	29 "
402	17 "	33 "	258	16 " 6 "	32 "	258	16 " 6 "	32 "
			287	18 "	35 "	287	18 "	35 "
16 feet deep 18 feet $7\frac{3}{16}$ inches high			14 feet deep 16 feet high			14 feet deep 12 feet $8\frac{3}{16}$ inches high		

Horse power	Class "E" Width		Horse power	Class "F" Width		Horse power	Class "N" Width	
	Single	Battery		Single	Battery		Single	Battery
99	7 ft.	13 ft.	207	9 ft.	17 ft.	173	7 ft.	13 ft.
124	8 "	15 "	241	10 "	19 "	216	8 "	15 "
148	9 "	17 "	276	11 "	21 "	260	9 "	17 "
173	10 "	19 "	310	12 "	23 "	303	10 "	19 "
198	11 "	21 "	345	13 "	25 "	347	11 "	21 "
223	12 "	23 "	379	14 "	27 "	390	12 "	23 "
247	13 "	25 "	414	15 "	29 "	433	13 "	25 "
272	14 "	27 "	448	16 "	31 "	477	14 "	27 "
297	15 "	29 "	483	17 "	33 "	520	15 "	29 "
321	16 "	31 "	517	18 "	35 "	564	16 "	31 "
346	17 "	33 "				607	17 "	33 "
371	18 "	35 "				650	18 "	35 "
16 feet 3 inches deep 15 feet 6 inches high			16 feet 9 inches deep 20 feet $11\frac{1}{4}$ inches high			18 feet 9 inches deep 24 feet $10\frac{3}{4}$ inches high		

ever, intermediate sizes between those in the table, and also a number of additional classes running principally to larger sizes. The widths given in the table are for a single boiler, and for two set in a battery. Space should be left at the rear and one side of

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the single boilers and at both sides of each battery for reaching the cleanout doors.

The smoke connections are taken from the top near the rear of the setting.

Worthington Boiler.—Fig. 10 shows the Worthington boiler with the outside casing removed. This is a water-tube boiler of the sectional class. It requires a relatively small space compared

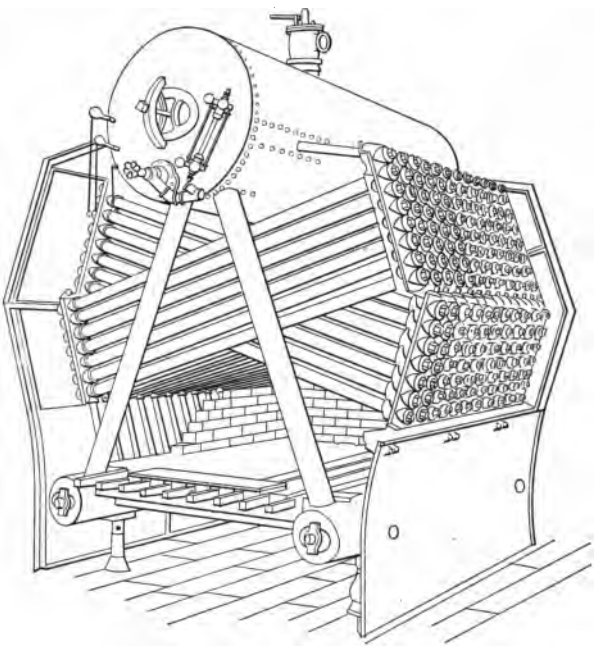

Fig. 10. Worthington Boiler.

with the amount of grate and heating surface. The furnace extends under the entire boiler and the whole interior construction is accessible for cleaning and repairs. Each tube is straight and is expanded into the headers at each end. The tubes are arranged directly over the furnace in oppositely inclined sections several tubes high. By this arrangement the whole heating surface is exposed to the full effect of the fire so that the movement of the water and the consequent evaporation is rapid. The water is fed into the large drum located at the top, and descends through the

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water legs, placed outside the furnace, to the mud drums at the base. It then passes through the tube connections into the lower headers, thence through the tubes over the fire into the upper headers, and then through the tube connections into the steam and water drums, where the steam is separated from the water and taken off at the top of the drum. The headers are made of steel

TABLE XVIII.

DIMENSIONS OF WORTHINGTON BOILERS.

END FIRED BOILERS.

Horse power.	Length over all.	Horse power.	Length over all.
	Width over all 6 ft. 8 in.; height to top of drum 8 ft. 4 in.		
	Ft. In.		Ft. In.
41	6 8	58	8 3
45	7 1	62	8 7
49	7 5	67	8 11
54	7 10		
	Width over all 8 ft. 9 in.; height to top of drum 10 ft. 1 in.		
75	7 11	109	9 10
84	8 4	117	10 3
92	8 10	125	10 9
100	9 4		

SIDE FIRED BOILERS.

	Width over all 8 ft. 9 in.; height to top of drum 10 ft. 1 in.		
42	6 0	117	10 3
59	6 11	125	10 9
84	8 4	142	11 9
100	9 4		
	Width over all 10 ft. 0 in.; height to top of drum 11 ft. 6 in.		
64	7 0	178	12 10
75	7 7	189	13 5
87	8 2	201	14 0
121	9 11	235	15 9
133	10 6	246	16 4
144	11 1	258	16 11

or iron, according to the service required, and are arranged in the boiler close together, forming complete side walls. Outside of the furnace, opposite the end of each tube, is a handhole of proper size to admit a tube or tube expander. The caps over the handholes are exposed for examination when the side doors of the casing are opened, and by removing these caps the inside of the tubes can be cleaned or inspected. The mud drums extend through the casing and are provided with handhole plates accessible from the outside. The only brick used in the setting except in the foundation walls, are those in the fire-brick lining of the furnace. The whole boiler, except the steam and water drum, is enclosed in a plate metal casing.

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Table XVIII. gives the over-all dimensions of some of the different sizes of the Worthington boiler. This is made in two patterns, called the "end-fired" and "side-fired." In the former the tubes run across the boiler, and space must be left at one side or the other for drawing the tubes. In the side-fired boiler they run lengthwise, and can be drawn from the front. When placed in a battery it is better to use the side-fired type, as with these it is not necessary to leave spaces between them for drawing the tubes.

Spaces should be left at the sides and rear of both single boilers and batteries for reaching the cleanout doors. The smoke connections are made at the top.

Cahall Vertical Boiler.—This is a water-tube boiler consisting of two drums arranged one above the other and connected with 4-inch tubes as in Fig. 11. The tubes are vertical and perfectly straight, and are expanded into the drums at each end. The upper or steam drum has an opening through its center for the discharge of the waste gases, which, in passing over the plates of the drum, give up part of their heat to the steam within, thereby tending to superheat it slightly. The water line in the upper drum is about 2 feet above the bottom of the drum; the drum itself being about 7 feet high inside. This gives a clear space of about 5 feet between the surface of the water and the steam outlet, which prevents the carrying over of water with the steam. An external circulating pipe connects with the upper drum just below the water level and is carried down outside the brickwork and connected with the lower drum just below the tube sheet. The fact that there is no steam in the circulating pipe causes the water which it contains to have a considerably greater specific gravity than that within the tubes; this condition produces a rapid circulation upward through the tubes and downward through the external pipe. The boiler rests upon four iron brackets riveted to the lower, or mud drum, which are in turn supported upon four piers of the foundation, thus carrying the entire structure without contact with the brickwork, and allowing freedom of expansion without straining the setting.

Table XIX. gives the over-all dimensions for this type of boiler. A space of about 18 inches should be left at the rear of a single boiler for the circulating pipe, but if two or more boilers

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are set in a battery, this should be increased to 36 inches or more for bringing out the blow-off pipes and reaching the cleanout doors. When two boilers are set in a battery the width is equal

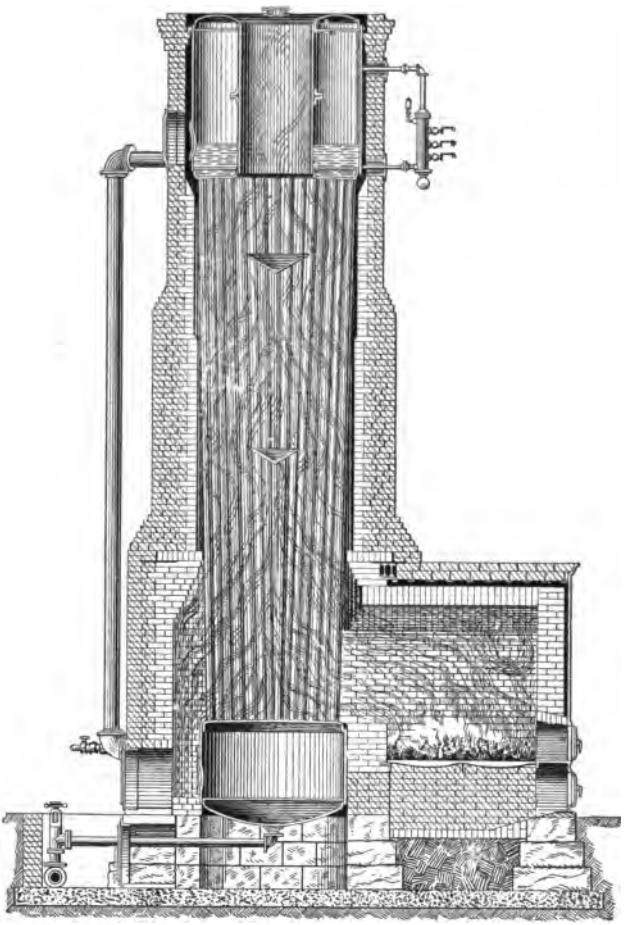

Fig. 11. Cahall Vertical Boiler.

to twice that of a single boiler, less 26 inches, and the width of an intermediate boiler in a battery of more than two is equal to that of a single boiler set alone, less 26 inches.

The smoke connections are taken from the top.

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The Figures in the Several Tables cover the extreme dimensions of the boilers when set. They are not intended to replace the exact dimensions, which should always be obtained from the makers for any particular case, especially if the available space is limited. They will, however, be found convenient for reference, and are sufficiently accurate for much preliminary

TABLE XIX.

DIMENSIONS OF CAHALL VERTICAL BOILERS.

Horse power.....	60	75	100	125	150	175	200	250
	Ft. In.	Ft. In.	Ft. In.	Ft. In.	Ft. In.	Ft. In.	Ft. In.	Ft. In.
Diam. of steam drum..	54	54	66	66	72	72	80	80
Diam. of mud drum....	42	42	54	54	60	60	68	68
Number of tubes.....	40	40	64	64	80	80	108	108
Diam. of tubes.....	4	4	4	4	4	4	4	4
Length of tubes.....	14	18	16	19	18	21	18	22
Length of setting.....	11 9	12 0	12 9	13 3	15 4	15 10	16 7	17 7
Width of setting.....	8 0	8 0	9 8	9 8	10 8	10 8	11 8	11 8
Height to steam outlet.	23 5	27 5	26 1	28 10	29 0	32 0	29 0	33 0

work. The dimensions given have been furnished by the makers or have been taken from published tables.

Cast-Iron Boilers for Heating

Cast-Iron Sectional Boilers.—There are many forms of cast-iron sectional boilers which are used principally for low-pressure heating in dwelling houses and other buildings of moderate size. The smallest of these are commonly made with a round grate and horizontal sections, while the larger ones are made up of vertical sections and have a grate rectangular in form. Sectional boilers of the latter design are of two general types, those in which the sections are connected directly by push nipples at top and bottom, and those having steam and return drums, as illustrated by the Gurney boiler shown in Fig. 12.

In this case, each section is independent of the others, except through the drums, a feature which tends to ensure a stable water line.

In using boilers of this type care must be taken not to use

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too many sections in a single boiler, for if the grate is made too long, the fire on the rear portion cannot be properly cared for, and the efficiency of the furnace will be impaired.

If more capacity is required, it is better to use two short boilers rather than a single long one. In general, the grate in this type

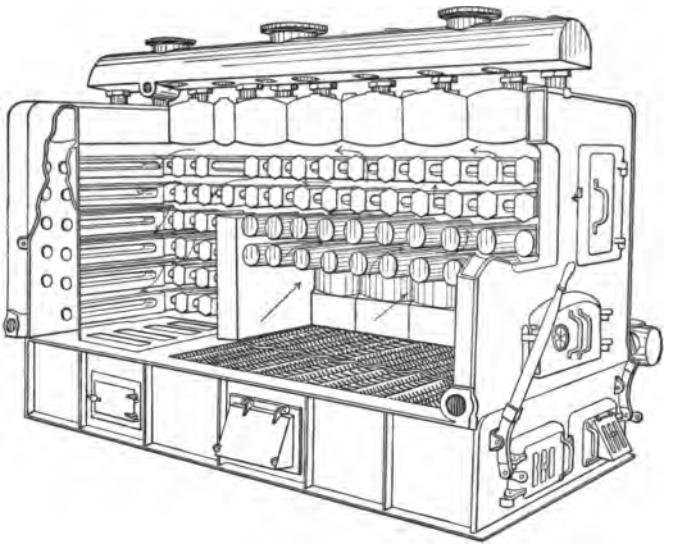

Fig. 12. Gurney Boiler.

of boiler should not be over 4 feet in length, or 5 feet at the most. In boilers of small size the grate usually extends to the last section, but for the larger sizes, with a good chimney draft, a bridge wall section may be used, as shown in Fig. 12, thus shortening the grate and increasing the ratio of heating to grate surface. When this is done, a higher rate of combustion is required to maintain the rated capacity. The usual setting for boilers of this type is simply a covering of some kind of non-conducting material like plastic magnesia or asbestos. The smaller cast-iron sectional boilers vary so greatly in size and shape

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for the same capacity that it is best to refer to the manufacturers' catalogues for dimensions.

Superheaters

Superheated steam has been defined as steam which has been heated to a temperature above that due to its pressure.

This is made practical use of in reducing cylinder condensation in reciprocating engines and also in connection with steam turbines, as described later.

Steam is superheated by passing it over surfaces raised to a high temperature, while out of contact with water.

The superheaters in common use are of two general types, known as the "separately fired" and "individual." The first of these is provided with a separate setting and is operated independently, while the later is installed in the same setting with the boiler and is practically a part of the boiler unit.

The individual superheater requires no special attendance, but is not so easily controlled as the separately fired apparatus, which is designed to do the work for the entire plant. The relation between the regular heating surface of the boiler and the superheating surface will depend upon the location of the latter and the temperature of the gases to which it is exposed. Assuming a furnace temperature of 2,500 degrees, and a flue gas temperature of 500 degrees surrounding the superheater, the relative proportions should be about as follows:

With the superheater placed nearer the furnace and exposed to a higher temperature, the amount of surface may be correspond-

Deg. of Superheat Required	Percentage of Total Heating Surface in Superheater
75	25
100	32
200	50

ingly reduced, but the *total* heating surface in the boiler and superheater, however, should remain practically the same as before.

The Heine superheater, attached to a boiler of the same make is shown in Fig. 13. It is placed above the front end of the steam

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drum and utilizes heat from the gases on their last pass before entering the smoke pipe. The heating surface is made up of bent tubes expanded into cast-steel headers and is connected with the

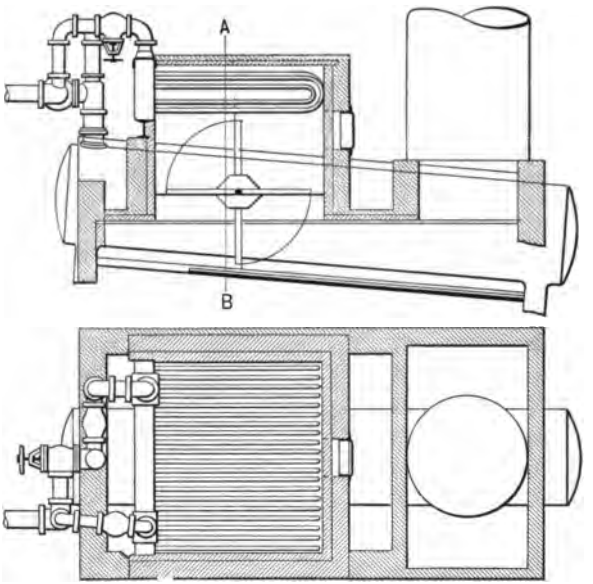

Fig. 13. Heine Superheater

steam outlet in such a manner that it may be by-passed when not in use, dampers being provided for cutting off communication with the furnace at the same time.

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CHAPTER VI.

DESIGN OF TUBULAR BOILERS

When water-tube boilers, or others of special form are to be used, the engineer has little to do with their design. Before selecting a boiler, however, he should satisfy himself as to the reliability of the manufacturers, make a careful examination of the extent and arrangement of the heating surfaces, and familiarize himself as much as possible with all the leading points in its design and construction. With boilers of this type the problem is one of selection rather than one of design.

When horizontal fire-tube boilers are to be used they should, if possible, be made to order from plans and specifications drawn up by the engineer, covering the more important points relating to their general proportions and construction. Space will not allow a detailed discussion of this subject, but a few simple rules and formulæ, together with various tables, will be sufficient to enable one to design a plain tubular boiler in all its essential parts, such as the size and number of tubes, type and dimensions of riveted joints, bracing, etc.

Materials.—Most boilers at the present time are made of "open hearth" steel. The usual requirements are that the plates shall have a tensile strength of not less than 55,000, nor more than 60,000 pounds per square inch of section, with not less than 56 per cent of ductility as indicated by construction of area at point of fracture; and on elongation of 25 per cent in a length of 8 inches.

The heads are made of "flange" steel, which is an extra quality of open-hearth steel, made with special reference to toughness and ductility.

The breaking load for mild steel in tension may be taken as 55,000 pounds per square inch, while the resistance to crushing in riveted joints is about 95,000 pounds.

Wrought-iron plates are rarely used except when especially called for. They are more ductile and require less care in working than steel, but otherwise have no advantage over steel plates,

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which are more easily obtained and stronger. The ultimate strength of wrought iron may be taken as 45,000 pounds per square inch for tension and 75,000 pounds per square inch for compression.

Rivets are made of iron or steel of a quality and strength similar to that used for the plates. They are subjected, under working conditions, to a shearing stress in addition to tension and compression. The ultimate resistance to shearing may be taken as 44,000 pounds for steel and 38,000 for wrought iron.

All stays and fastenings which require welding should be of the best quality of wrought iron. Where welding is not necessary mild steel may be used instead.

Manhole frames and covers are usually made of cast iron or cast steel, although wrought iron and mild steel are often used for this purpose. The finer grades of cast iron commonly known as "gun iron" have a tensile strength varying from 20,000 to 25,000 pounds; the resistance to crushing is about three times as great for small blocks, while somewhat less for larger pieces.

Factor of Safety.—In designing the different parts of any structure, a certain allowance in strength must be made to overcome any flaw which may be present in the material, and to offset any sudden load which may be thrown upon the structure when in use. This allowance is called a *factor of safety* and varies according to the material used and the kind of load to be supported. A factor of safety of 6 is recommended for tubular boilers in general use, although 5 is usually considered safe for well constructed boilers of good design.

Drilling and Punching Plates.—It is usual in the best class of work to drill the rivet holes after the plates are in place. When the plates are punched, the metal between the holes is weakened. Plates that are first punched and then rolled to form are often distorted so that the holes come unfair for riveting; this defect is avoided when the holes are drilled with the plates in position.

General Proportions

The diameter and general proportions of a boiler are determined by the horse power required, the pressure to be carried, and to some extent by the available space in the boiler room. As the shell plates in this type of boiler are exposed directly to the fire

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they should not in general be over ½ inch in thickness, which limits the diameter for any given pressure. Another factor bearing upon the diameter is the tube area; this is usually made at least one seventh of the grate area.

Tubes.—The sizes of tubes most commonly used are those having outside diameters of 2½, 3, 3½, and 4 inches. A common rule for determining the diameter of tubes to be used for any given length of boiler is to allow 1 inch in diameter for each 5 feet in length for anthracite coal, and for each 4 feet in length for bituminous.

The number and arrangement of the tubes is a matter of much importance; they should not be crowded together too closely as increasing the number beyond a certain limit decreases their efficiency as heating surface. There should be a clear space of 1 inch between the tubes both vertically and horizontally, with a vertical space of 2 inches at the center. Tubes should not be placed nearer than 3 inches to the shell, owning to the liability of grooving, due to unequal expansion of shell and tubes. A clear space of 6 inches or more should be left between the tubes and the bottom of the shell to serve as a mud drum.

Table VII. gives a suitable number of tubes for boilers of different diameters, as well as other useful data in connection with boiler design.

Steam Space.—In heating boilers it is customary to make the steam space about one third of the contents of the shell after deducting the space occupied by the tubes. For power boilers it is often proportioned according to the amount of steam required. A method commonly employed is to make it equal to the volume of steam used by the engine in 20 seconds. The normal water line should be from 4 to 6 inches above the upper row of tubes; 5 inches is usual for boilers 54 and 60 inches in diameter.

Strength of Shell

Thickness of Shell.—The relations between the thickness of shell, steam pressure, and diameter of boiler, are expressed in the following equations:—

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$$t = \frac{d \times p \times F}{2 \times S \times E}$$

$$p = \frac{2 \times S \times t \times E}{d \times F}$$

$$d = \frac{2 \times S \times t \times E}{p \times F}$$

in which

t = thickness of shell plates, in inches.

p = steam pressure, in pounds per sq. inch.

d = diameter of boiler, in inches.

F = the factor of safety.

S = the ultimate strength of the shell plates, in pounds per sq. inch.

E = the efficiency of the riveted joints.

The following may be taken as the average values of E for different types of riveted joints:—

Single riveted lap joint, 55 per cent.

Double riveted lap joint, 70 per cent.

Triple riveted lap joint, 75 per cent.

Double riveted butt joint, 80 per cent.

Triple riveted butt joint, 87 per cent.

Example.—What thickness of plate will be required for a boiler 60 inches in diameter, of mild steel with double riveted butt joints, to carry a pressure of 100 pounds, with a factor of safety of 5?

$$t = \frac{60 \times 100 \times 5}{2 \times 55{,}000 \times 0.80} = 0.341$$

which is a little less than 3/8 inch. In practice it is customary to add at least 1/16 inch to offset the effects of corrosion, which would make the required thickness of the shell in the above case, $3/8 + 1/16 = 7/16$ inch.

Table XX. gives the working pressures for boilers of different diameters, thickness of shell, and type of joints, with varying factors of safety. In using this table in the design of new boilers, an additional amount must be added to offset the effects

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of corrosion as already stated. Boiler heads are usually made from 1/16 to ⅛ of an inch thicker than the shell plates.

Riveted Joints.—For all ordinary work it is more convenient to use proportions for the various types of riveted joints which

TABLE XX.

WORKING PRESSURES FOR TUBULAR BOILERS.

Assumed strength of plates, 55,000 pounds; assumed efficiency of joints, single lap 55 per cent, double lap 70 per cent, double butt 80 per cent, triple butt 87 per cent.

Diameter of boiler, inches	Thickness of shell, inches	Factor of safety 6				Factor of safety 5				Factor of safety 4			
		Kind of joint				Kind of joint				Kind of joint			
		Single lap	Double lap	Double butt	Triple butt	Single lap	Double lap	Double butt	Triple butt	Single lap	Double lap	Double butt	Triple butt
30	¼	82	107	123	137	98	128	147	163	123	160	184	204
	5/16	103	134	155	172	124	161	186	205	155	201	232	257
	⅜	124	161	186	205	148	192	221	245	185	240	277	307
	7/16	145	188	217	240	173	225	259	286	216	281	324	358
	½	165	215	248	275	198	257	296	328	247	321	370	410
36	¼	69	90	103	115	82	107	123	137	103	134	154	171
	5/16	86	112	129	143	103	134	154	171	129	168	193	214
	⅜	103	134	155	171	123	160	185	204	154	200	231	255
	7/16	120	157	181	201	144	187	216	239	180	234	270	299
	½	138	180	207	229	165	214	247	274	206	268	309	342
42	¼	59	76	88	98	70	91	105	117	88	114	132	146
	5/16	73	96	110	122	88	115	132	145	110	143	165	182
	⅜	88	115	133	147	105	137	158	175	132	171	198	219
	7/16	103	134	155	170	123	160	185	204	154	200	231	255
	½	118	154	178	197	142	184	212	235	177	230	265	294
48	¼	52	67	77	86	62	80	92	102	77	100	115	128
	5/16	64	84	96	106	77	100	115	127	96	125	144	159
	⅜	78	100	116	129	93	120	139	154	116	150	174	193
	7/16	90	118	135	150	108	140	162	179	135	176	203	224
	½	104	135	155	172	124	162	185	205	155	202	232	257
54	¼	46	59	68	76	54	70	82	90	68	88	102	113
	5/16	58	75	86	96	69	90	103	114	86	112	129	143
	⅜	69	90	103	115	82	107	123	137	103	134	154	171
	7/16	80	105	120	134	96	125	144	159	120	156	180	199
	½	92	119	137	152	109	142	163	182	137	178	205	227
60	¼	41	53	62	68	49	64	73	82	62	80	92	102
	5/16	51	67	77	86	62	81	93	102	77	100	116	128
	⅜	62	81	93	102	74	96	110	122	93	120	138	153
	7/16	72	94	109	120	87	112	129	143	108	140	162	179
	½	82	108	124	137	99	128	148	164	124	160	185	205
66	¼	38	49	56	62	45	58	67	74	56	73	84	93
	5/16	47	61	70	78	56	73	84	93	70	91	105	116
	⅜	56	73	84	92	67	87	100	113	84	109	126	139
	7/16	66	85	98	109	78	102	118	130	98	127	147	163
	½	75	98	113	125	90	117	134	149	112	146	168	186
72	¼	35	45	51	57	41	54	61	68	52	67	77	85
	5/16	43	56	65	71	51	67	77	85	65	84	96	107
	⅜	51	67	77	85	61	80	92	102	77	100	116	127
	7/16	60	78	91	101	72	94	108	119	90	117	135	145
	½	69	90	104	114	82	107	123	137	103	134	155	171

experience has proved to be satisfactory than to prepare new designs. Tables XXI. and XXII. give the different dimensions for the more common forms of riveted joints as recommended

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by the Hartford Steam Boiler Inspection and Insurance Company. In giving the pitch of the rivets for the girth seams, they are assumed to be of the same diameter as in the longitudinal seams.

TABLE XXI.*

DOUBLE AND TRIPLE RIVETED LAP JOINTS.

Thickness of plate.	Diameter of rivets.	Pitch of rivets.	Distance between rows of rivets.	Edge of plate to center of rivets.	Pitch of girth seam rivets.	Efficiency of joint.
Double riveted.						
$\frac{1}{4}$	$\frac{11}{16}$	$2\frac{7}{8}$	$1\frac{11}{16}$	$1\frac{1}{8}$	$2\frac{1}{16}$	74%
$\frac{5}{16}$	$\frac{3}{4}$	$2\frac{3}{4}$	$1\frac{11}{16}$	$1\frac{7}{32}$	$2\frac{1}{8}$	72%
$\frac{3}{8}$	$\frac{7}{8}$	$3\frac{1}{4}$	$2\frac{3}{16}$	$1\frac{13}{32}$	$2\frac{3}{8}$	70%
$\frac{7}{16}$	$\frac{15}{16}$	$3\frac{1}{4}$	$2\frac{3}{16}$	$1\frac{9}{16}$	$2\frac{7}{32}$	70%
$\frac{1}{2}$	1	$3\frac{1}{4}$	$2\frac{1}{16}$	$1\frac{17}{32}$	$2\frac{1}{4}$	68%
Triple riveted.						
$\frac{1}{4}$	$\frac{3}{4}$	3	2	$1\frac{1}{16}$	$2\frac{1}{16}$	77%
$\frac{5}{16}$	$\frac{15}{16}$	$3\frac{1}{4}$	$2\frac{1}{16}$	$1\frac{1}{8}$	$2\frac{1}{8}$	76%
$\frac{3}{8}$	$\frac{7}{8}$	$3\frac{1}{4}$	$2\frac{3}{16}$	$1\frac{7}{32}$	$2\frac{1}{8}$	75%
$\frac{7}{16}$	1	$3\frac{1}{2}$	$2\frac{1}{16}$	$1\frac{13}{32}$	$2\frac{3}{16}$	75%
$\frac{1}{2}$	$1\frac{1}{16}$	$3\frac{1}{16}$	$2\frac{1}{8}$	$1\frac{9}{16}$	$2\frac{1}{4}$	75%

* From Ryerson's *Monthly Journal.*

Girth seams are subjected to a stress due to the pressure upon the heads. It can be shown by a simple process, that this stress for any given length of seam is just one half that upon an

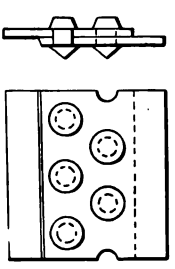

Fig. 14. Double Riveted Lap Joints.

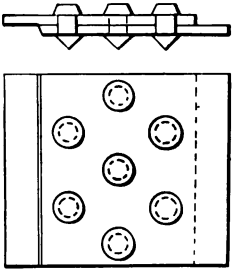

Fig. 15. Triple Riveted Lap Joints.

equal length of a longitudinal seam. A single riveted lap joint of usual dimensions has an efficiency of 50 per cent or more, so that

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if all girth seams are made of this type, the boiler will be as strong relatively as the solid plates, in a direction parallel with its axis.

TABLE XXII.*

DOUBLE AND TRIPLE RIVETED BUTT JOINTS.

Thickness of plate.	Diameter of rivets.	Pitch of rivets.	Width of outside butt strap.	Width of inside butt strap.	Thickness of butt straps.	Distance between rows of rivets.	Edge of butt straps to center of rivets.	Pitch of girth seam rivets.	Efficiency of joint.
Double riveted.									
5/16	11/16	2 1/2 x4 1/2	4 1/2	9	1/4	2 1/2	1 1/4	2 1/2	83%
3/8	3/4	2 5/8 x4 3/4	4 5/8	9 1/4	5/16	2 7/8	1 7/16	2 5/8	83%
7/16	13/16	2 13/16x4 7/8	5 1/16	10 1/2	3/8	2 5/8	1 11/16	2 1/2	82%
1/2	7/8	2 7/8x5 1/8	5 5/8	11 1/4	7/16	2 13/16	1 13/16	2 1/2	80%
Triple riveted.									
5/16	11/16	3 1/4x6 1/2	9 1/2	14	1/4	2 1/8	1 1/4	2 1/8	88%
3/8	3/4	3 1/2x6 3/4	9 3/4	14 1/4	5/16	2 3/8	1 3/8	2 1/8	87.5%
7/16	13/16	3 3/4x6 3/4	10 1/8	15 5/8	3/8	2 1/2	1 7/16	2 7/8	86%
1/2	7/8	3 7/8x7 1/4	11	16 3/4	7/16	2 5/8	1 1/2	2 1/2	86.6%

* From Ryerson's *Monthly Journal*

While lap joints are amply strong for the pressures indicated, when new, experience has shown that they have a tendency to become weakened through extended use, due to a slight bending

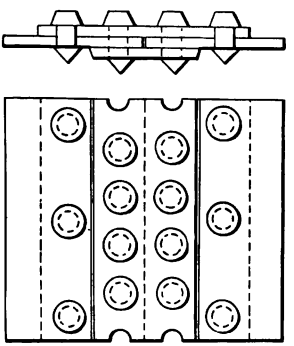

Fig. 16. Double Riveted Butt Joints.

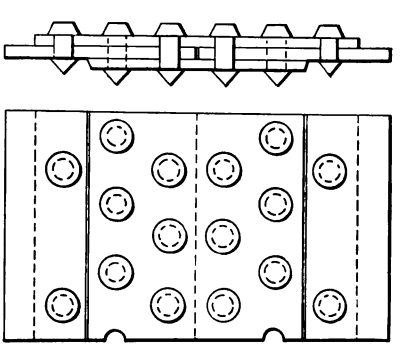

Fig. 17. Triple Riveted Butt Joints.

strain when under tension. For this reason it is better to use butt joints in all cases, unless it be for heating boilers carrying pressures less than 10 pounds, and even under these conditions the butt joint is preferable.

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Boiler Bracing

The surface to be supported by the braces is evidently that portion of each head above the tubes. In practice, however, it is not necessary to put in braces enough to sustain the whole of this area, for the reason that when the head is flanged and riveted to the shell a portion of it becomes stiff enough to carry the

Fig. 18

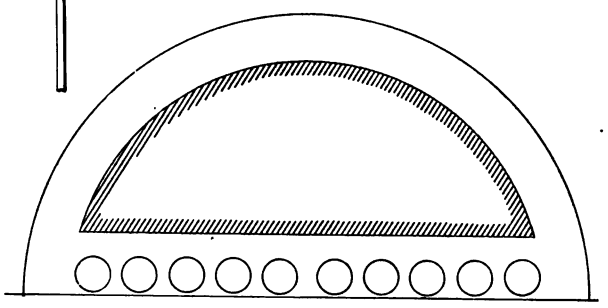

Fig. 19

Figs. 18 and 19. Indicating Part of Head Requiring Bracing.

pressure without depending upon the braces. The distance that becomes self-supporting may be determined by the formula

$$D = \frac{4{,}484 \times T}{10 \times p} + 0.5$$

in which $D =$ the distance inward from the shell as indicated in Fig. 18, $T =$ the thickness of the head, and $p =$ the steam pressure per square inch.

When a boiler head has several rows of tubes expanded into it, a portion of the plate for a short distance above the upper row is sufficiently supported by the tubes and it is usually safe to call this distance two thirds of that determined by the above formula. This leaves an area of the shape and location indicated by the shaded portion in Fig. 19 to be supported by the braces. Multiplying this area in square inches by the steam pressure per square inch gives the total load to be carried, and the size and number of braces must be proportioned accordingly.

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Computing Area of Segment.—As a matter of convenience, the rule is given here for computing the area of the segment of a circle, which is necessary in finding the area of the shaded portion in Fig. 19.

First compute the area of the sector *o a b c* (Fig. 20), which is equal to the distance *a b c* measured around the shell, mul-

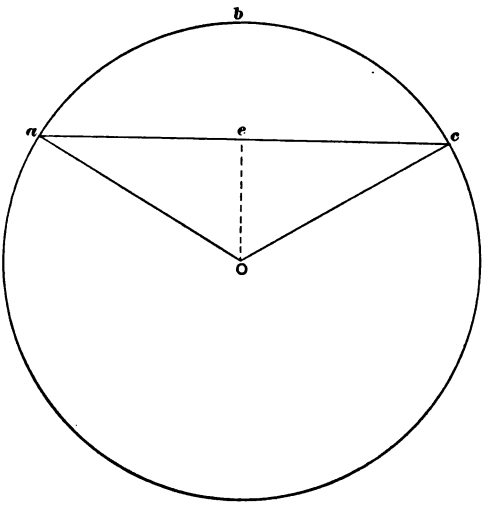

Fig. 20.

tiplied by one half the radius *o a*. Next compute the area of the triangle *o a e c*, which is equal to *a e* multiplied by *o e*, and subtract this from the area of the sector first found. The result will be the area of the segment. In making this computation the dimensions can be scaled from the drawing with sufficient accuracy.

Points of Support.—The maximum distance between the points of support, that is, the distance between the rivets fastening the head to the stays or to the stiffening bars, may be computed by the formula

$$p = \sqrt{\frac{C \times t^2}{P}}$$

in which

p = the maximum distance between supports in inches.

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t = thickness of head, in sixteenths of an inch.

P = steam pressure in pounds per square inch.

C = 112 for plates 7/16 inches and under and 120 for plates over 7/16 inches in thickness.

Assuming a steam presure of 100 pounds per square inch, we find the values of p to be as follows for different thicknesses of plate.

Table XXIII.

Distances Between Rivets for Different Thicknesses of Heads.

"t"	"p" Computed as above.	Values used in practice.
In.	In.	In.
7/16	6.3	4
1/2	7.3	4½
9/16	8.7	5½
5/8	9.8	6
11/16	10.9	6½

As the formula gives maximum values, it is customary to reduce the results somewhat in practice, so that the last column represents more nearly the actual distances used. There are two methods commonly employed for staying the heads of horizontal tubular boilers, known as "through bracing" and "crow-foot bracing."

Through Bracing.—Through bracing consists of rods passing through the entire length of the boiler with heavy nuts and washers at the ends on the outside, and checknuts on the inside. Stiffening pieces of channel or angle iron are riveted to the heads to support them. Fig. 21 shows the general arrangement of the channel or angle irons and the ends of the rods or braces, together with a section of the stiffening bars and head. The upper bar in this case consists of a channel iron with an angle riveted to its upper flange to give greater strength, while the lower bar is made up of two angles riveted to the head as shown. Where angles are used instead of channel iron, it is necessary to countersink slightly around the holes for the braces, in order to give a solid bearing for the washer which is placed under the checknut.

The outside washer is sometimes made of copper, slightly cup-shaped on the under side in order to make a tight joint around the edge. If cast-iron washers are used they should be

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embedded in red lead. When channel bars will give the required strength they are preferable to angles for stiffening pieces, but for high pressures the standard weights will in many cases be

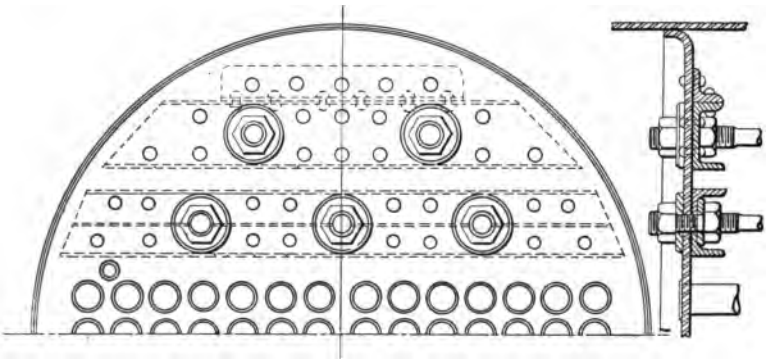

Fig. 21. Arrangement of Through Braces.

found too light, and resort must be had to angle irons, of which there is a great variety to choose from. In this form of bracing the channel bars act as beams carrying the load due to the pull on the rivets, and the beams in turn are supported by the rods.

Proportions for Through Bracing.—Through bracing is commonly employed in boilers 66 and 72 inches in diameter; five

TABLE XXIV.

SIZES OF BRACES AND STIFFENING BARS FOR 66- AND 72-INCH BOILERS.

	In.	In.
Diameter of boiler	66	72
Distance from center of upper row of tubes to center of lower channel	7½	8½
Distance between centers of channels	9	10
Distance between lower braces	15	16½
Distance between upper braces	19	20
Channel bars, inches	7x2.67x0.57	Two 3½x2½x½ angles*
Angle, inches	3½x3x½*	3½x3x½*
Diameter of braces	1⅜	1⅜
Ends upset to	2¼	2¼
Diameter of channel rivets	1	1
Diameter of cast iron washers	6	6
Thickness of cast iron washers	1	1

* Longer flange riveted to head.

braces being used and arranged as shown in Fig. 21. The distance between the upper braces should not be less than 17 or 18 inches owing to the difficulty of access for inspection if placed nearer together.

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Table XXIV. gives the sizes of braces and stiffening bars for boilers 66 and 72 inches in diameter, having the number of tubes given in Table VII. Each boiler has five braces of mild steel proportioned to carry a pressure of 100 pounds per square inch with a factor of safety of 6. In this table, provision has been made for a reasonable amount of corrosion. The stiffening bars are usually spoken of as "channels" regardless of their makeup. The "angle" mentioned in the table is the angle iron riveted to the upper channel. The ends of the braces should be upset to

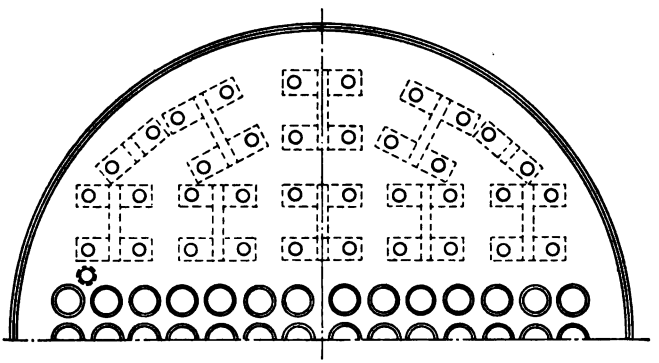

Fig. 22. Arrangement of Crow-foot Braces.

such a size that the diameter at the root of the thread shall be equal to, or slightly greater than the body of the brace.

Crow-foot Bracing.—In this form of bracing the supported points are attached to a forging known as a "crow-foot." (See Fig. 22.) A diagonal stay with a forked head and pin is used to connect each crow-foot with the boiler shell, to which it is fastened with two or more rivets. A crow-foot joined to the boiler head with four rivets as in Fig. 23 is called a double crow-foot; when only two rivets are used as in Fig. 24 it is called a single crow-foot. As the crow-foot and connecting rod are forgings, they are necessarily made of wrought iron containing welded joints, and should not carry a load of more than 5,000 pounds per square inch of section, measured at the weakest point.

Solid Steel Braces are now replacing the older form of crow-foot brace just described. These being of one piece, without

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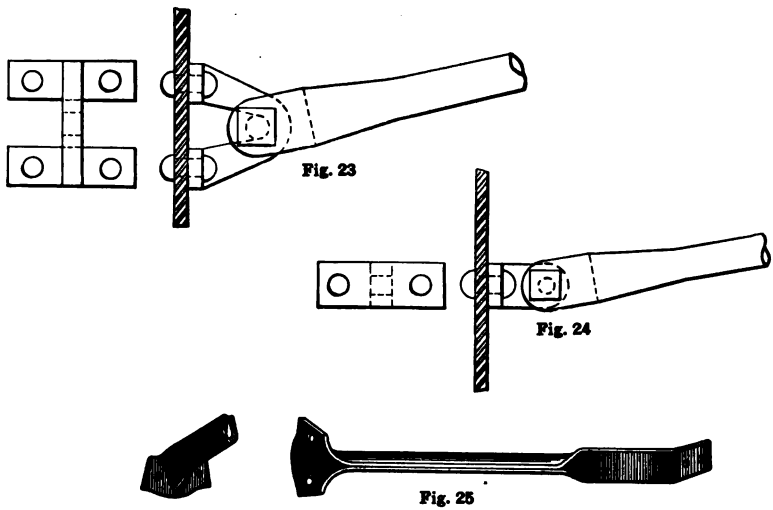
welds, can be made of steel and therefore have a much greater tensile strength per square inch of section. A load of 10,000 pounds per square inch may be safely carried, as against 5,000 pounds with a welded brace. A common form is shown in Fig. 25. These braces are attached to the head and shell with two rivets in each end. The rivets connecting the brace with the



Figs. 23, 24 and 25. Forms of Crow-foot Braces.

head are intension and those connecting it with the shell are subjected to a shearing stress and should be proportioned accordingly. No rivet, however, used in connection with braces of any kind should be less than ¾ inch in diameter.

Solid steel braces are commonly made of such size that the weakest part shall have a sectional area of at least one square inch. Two 1-inch rivets in each end of a brace of this size will give the connections equal strength with the body of the brace.

Proportioning Diagonal Bracing.—As the brace is carried back diagonally from the head to the shell, the load which it supports perpendicular to the head is less than that along the axis of the brace. The angle which the brace makes with the shell should not in general be much over 20 degrees, in which case the supporting force at the head is only about 0.9 of that carried by the brace in the direction of its length.

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In laying out the braces for a boiler we first find the area of the segment to be supported and multiply it by the steam pressure per square inch to find the total load. This divided by the "working strength" of the type of brace to be used will give the number required. These should be arranged in the general manner shown in Fig. 26, and the distance between the rivets checked

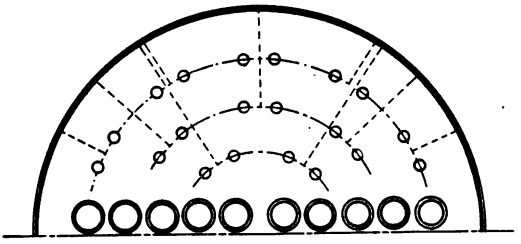

Fig. 26. Layout of Rivets for Crow-foot Braces.

by the values given in Table XXIII. If p is found to be too great, more braces should be added. Crow-foot braces are best adapted to low pressures and to boilers of moderate diameters, while through bracing is better for the larger sizes when high pressures are to be carried. Fig. 26 shows the general arrangement for crow-foot bracing, and Table XXV. gives

TABLE XXV.

DATA FOR CROW-FOOT BRACING.

Diameter of boiler	42 in.	48 in.
Rows of crow-foot braces	2	3
Number in 1st row	4	5
Length of braces in 1st row	42 in.	42 in.
Number in 2d row	2	3
Length of braces in 2d row	48 in.	48 in.
Number in 3d row		2
Length of braces in 3d row		60 in.
Distance from shell to 1st row	6 in.	6 in.
Distance between rows	5 in.	5 in.
Least area of brace in weakest part (solid braces).	0.8 sq. in.	0.8 sq. in.
Least diameter of rivets	$1\frac{3}{8}$ in.	$1\frac{3}{8}$ in

data and dimensions for boilers 42 and 48 inches in diameter, braced in this way for a working pressure of 100 pounds, with a factor of safety of 6.

A good arrangement for boilers 54 and 60 inches in diameter is a combination of "through" and "diagonal" bracing, using the lower channel and through braces as shown in Fig. 27 and re-

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placing the upper channel and through braces with those of the solid crow-foot type. This arrangement makes it much easier to

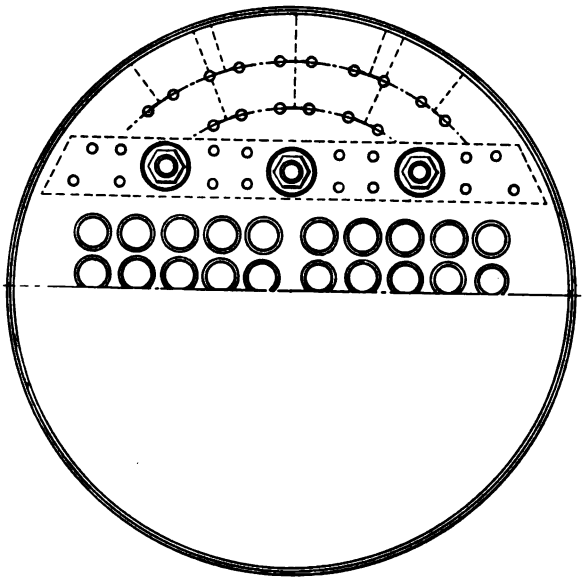

Fig. 27. Arrangement for Combined Through and Diagonal Bracing.

TABLE XXVI.

DATA FOR COMBINED THROUGH AND DIAGONAL BRACING.

Diameter of boiler	54 in.	60 in.
Distance from center of upper row of tubes to center of channel	7½ in.	8 in.
Distance between through braces	12¼ in.	12¼ in.
Channel bar, inches	6x2 39x52	6x2 39x52
Diameter of braces	1¼ in.	1¼ in.
Ends upset to	2 in.	2 in.
Diameter of channel rivets	⅞ in.	⅞ in.
Diameter of cast iron washers	5 in.	5 in.
Thickness of cast iron washers	1 in.	1 in.
Rows of crow-foot braces	2	2
Number in 1st row	5	5
Length of braces in 1st row	42 in.	42 in.
Number in 2d row	2	3
Length of braces in 2d row	48 in.	48 in.
Distance from shell to 1st row	6 in.	6 in.
Distance between rows	5 in.	5 in.
Least area of brace in weakest part (solid braces)	0.8 sq. in.	0.8 sq. in.
Least diameter of rivets	⅞ in.	⅞ in.

enter a boiler for inspection, which is difficult if not impossible when complete through bracing is used in boilers of small size.

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Table XXVI. gives data for the combination bracing of boilers 54 and 60 inches in diameter, for a working pressure of 100 pounds per square inch, with a factor of safety of 6.

Low Pressure Boiler.—In designing a boiler for low-pressure heating it is good practice to give it sufficient strength to safely carry 100 pounds pressure; this allows a large factor of safety

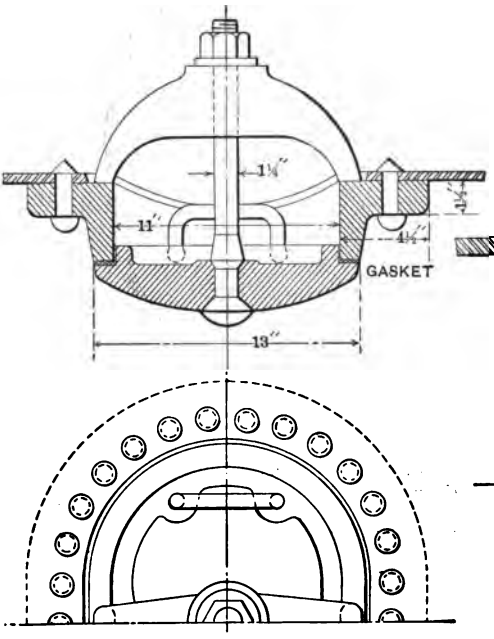

Fig. 28. Cast-Iron Manhole Ring and Cover.

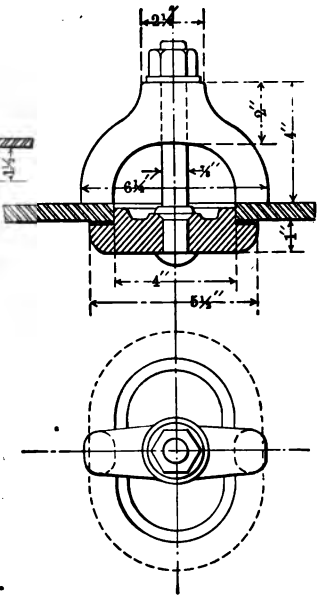

Fig. 29. Handhole Cover.

and makes ample provision for the effects of corrosion and adds to the life of the boiler.

Nozzles, Brackets, Feed Pipes, Etc.

Manhole.—Access to the interior of a boiler is provided for by a manhole; usually located in the top of the shell, and must be large enough for a man to pass through easily. The opening is oval in form and commonly 11 by 15 inches in size. Sometimes the manhole is placed in the front head of a boiler below the tubes. If this is done additional stays must be provided to take the place of the tubes which are necessarily omitted.

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Fig. 28 shows a common form of cast-iron manhole cover and frame, designed to carry a pressure of 150 pounds per square inch. Fig. 30 shows a section through a forged iron or steel manhole, designed for the same pressure.

Handholes.—These are placed in the lower part of each head for use when washing out the boiler. Fig. 29 shows a common form, consisting of a gun iron or cast steel cover and yoke. A gasket is provided for making a tight joint the same as in the

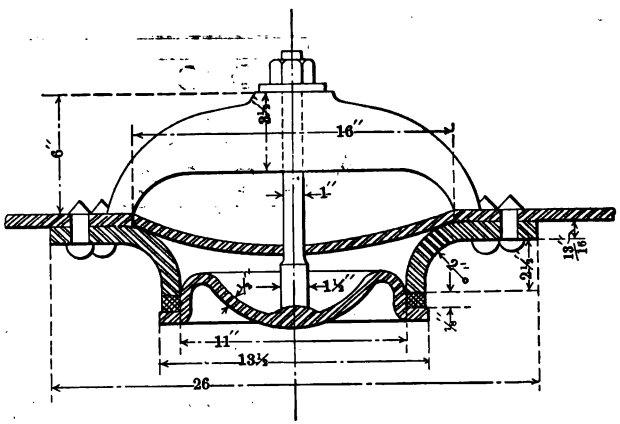

Fig. 30. Steel Manhole Ring and Cover.

case of a manhole. The bearing surface should be at least ¾ inch wide for a handhole and 1 inch for the manhole.

Nozzles.—These are made of the same material as the manhole, either gun iron or cast steel. They are usually of the same diameter throughout, but for power work it is better to taper them, making the end connecting with the boiler somewhat larger than the size called for. This form presents less resistance to the flow of steam than when the diameter is made uniform.

Brackets.—Horizontal tubular boilers are usually supported by cast-iron brackets riveted to the shell and resting upon the brickwork of the setting. The brackets are so placed that their bearing surfaces are 3 to 4 inches above the center line of the boiler. Boilers up to 17 feet in length are provided with two pairs of brackets, while those of greater length should have a third pair located midway between the others. Care must be taken to locate the brackets so that the load shall be evenly distributed among

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the entire number. Figs. 31 and 32 show the usual forms of brackets now in use. The former is the more common, while the latter has the advantage of greater strength.

Boiler makers have standard patterns for manholes, nozzles, brackets, etc., which they use with boilers of different sizes; so in specifying, it is usually best to rely upon forms and sizes which their experience has found to be satisfactory, rather than to pre-

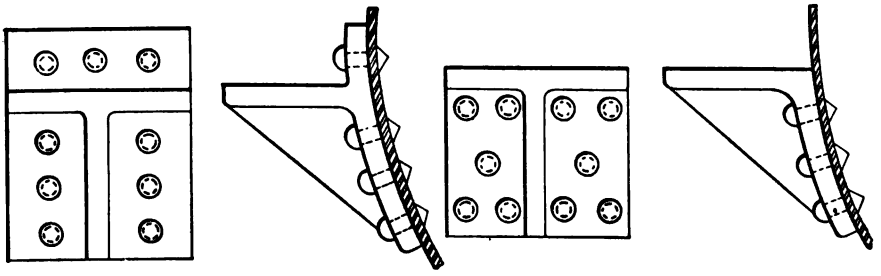

Fig. 31. Fig. 32.

Supporting Brackets.

pare special patterns. The examples given here will be found useful for comparison and also an assistance in the design of special forms.

Feed Pipe.—The feed pipe should be of brass, except that portion exposed to the hot gases in the extension front, which should be of extra heavy wrought iron. The pipe usually enters the front head just above the tubes at one side and is carried to a point about 2 feet from the rear head, then across, and ends in an elbow looking down between the tubes and the shell. Sometimes the end is plugged and small holes are drilled in the sides of the pipe. The object in any case is to deliver the water in the coolest part of the boiler, and in such a manner as to prevent its being discharged in any amount against the hot plates.

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CHAPTER VII.

BOILER FURNACES

The requirements of a good furnace are: first, ample capacity, both as regards grate surface, and space for combustion; second, grates of such form that ashes and clinkers may be easily and quickly removed; third, the construction should be such that the gases may be completely burned before coming in contact with the comparatively cool surfaces of the boiler shell and tubes; fourth, thick or hollow walls should be provided to prevent the loss of heat by radiation The furnace will necessarily vary in shape and size with the type of boiler and the kind of fuel to be used. With anthracite coal the grates may be placed from 24 to 30 inches from the boiler shell, but if bituminous coal is to be used this distance should be from 36 to 48 inches for the average grades and preferably more for fatty or gaseous coals. There should be a uniform and ample supply of air to the ash pit, and in the case of smoky fuels, provision should be made for admitting a small quantity of air above the fire for a short time after adding fresh coal.

Grates

Proportions.—Grates should not be larger than can be conveniently fired and cleaned. A hand-fired grate should not in general be over 6 feet in length, and if over 4 feet in width, two firing doors should be provided. The grates are usually inclined from ¾ to 1 inch per foot in length toward the rear, to aid in firing.

The form of grate bars and the amount of air space depend upon the fuel to be used. For fine sizes of anthracite the air spaces should be as narrow as can be used without choking the draft, and are often made as small as ¼ inch or even less. For the larger sizes of anthracite and for bituminous or coking coal the spaces are sometimes increased to 1 inch. In general, the free area through the grate varies from 30 to 50 per cent of the total area.

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Plain or Stationary Grates.—These are usually made up in sections of from one to three bars and should not be over 3 feet in length. They should be thinner at the bottom than at the top, to prevent clogging with ashes and cinders.

Grates of this kind are supported upon cross bars built into the brickwork of the setting.

The "herring bone" grate bar, Fig. 33, is another form in common use. In plants of small size, and especially those burning anthracite coal of egg size and larger, the plain grate is probably as satisfactory as any.

Shaking Grates.—For coals which are high in ash, and those which tend to form clinker, shaking or rocking grates are pre-

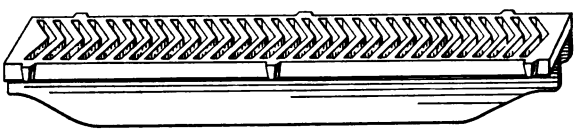

Fig. 33. Herring Bone Grate Bar.

ferable. There are many forms of these grates, by means of which the clinker can be broken up without opening the furnace doors. The advantages claimed for shaking grates are: ease of operation, the avoidance of admitting cold air to the furnace through open doors, and the thoroughness with which the clinker may be broken up and the ashes removed from the fire. To be effective, the construction should be such that the moving parts will not clog, and the clinker will be thoroughly and easily broken up by their action.

Grates of this kind may be divided into three general classes: the shaking or rocking grate, the dumping grate, and the combined shaking and dumping grate. The Foster grate with part of its bars removed is shown in Fig. 34. This is a typical shaking grate, and is made up of flat bars supported at each end upon cross pieces which in turn rest upon the side bars of the frame. There are two cross supports or "rocking bars" at each end, and alternate grate bars rest upon these two sets of supports. The grate has a direct lateral or sliding movement and is so fitted that there is no lost motion. One of the shakers operates one half of the bars and the other the remainder. The grate presents a level surface at all times as shown in the cut.

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In the dumping grate the sections are made larger, and when the fires are to be cleaned from clinker a section is rocked from

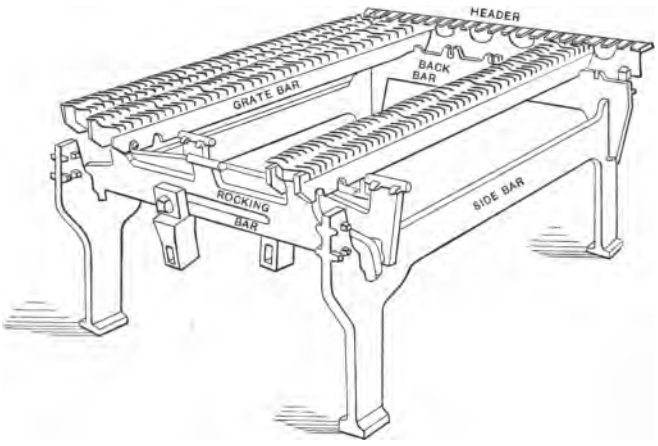

Fig. 34. Foster Shaking Grate.

a horizontal to a vertical position, thus breaking up the clinker and allowing it to fall through the large opening thus made,

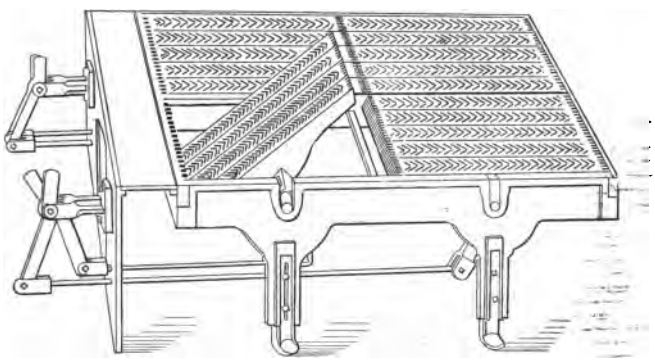

Fig. 35. Salamander Dumping Grate.

into the ash pit. The Salamander grate, Fig. 35, illustrates this type.

The McClave rocking and dumping grate, Figs. 36 and 37, illustrates the third type. Fig. 36 shows the rocking motion

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which is used to break up a soft coal fire when it cakes, or remove the fine ashes from a hard coal fire when there is but little clinker formed. The cut-off movement in Fig. 37 is for breaking up large clinkers and dumping them into the ash pit. The grates are divided into two parts so that each may be operated independently of the other.

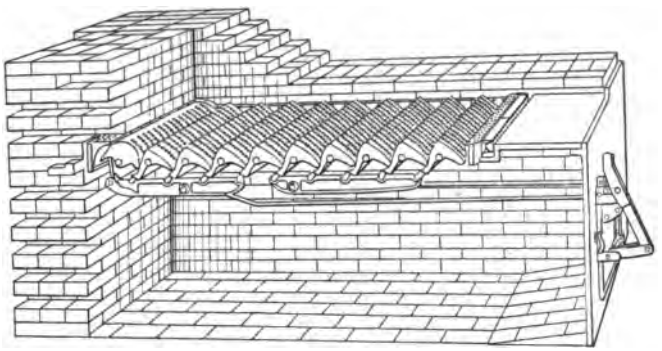

Fig. 36. McClave Rocking and Dumping Grate.

Fig. 38 shows the White hollow-blast rocking grate. This is designed especially for burning refuse and low-grade fuel. The

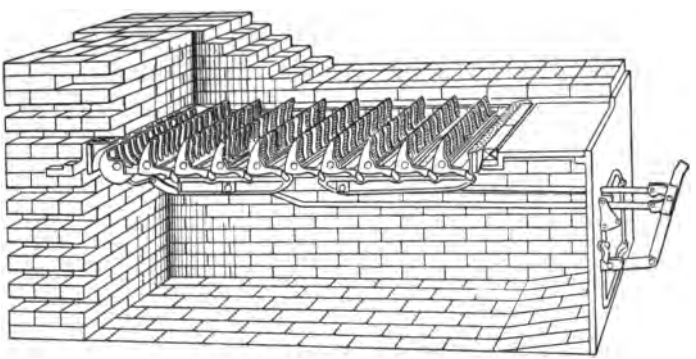

Fig. 37. McClave Grate.

furnace is so constructed that the air from the blower passes into the bridge-wall duct, then by means of hollow trunnions into air chambers running the full length of each grate bar, from which it passes upward in finely divided streams through the tops

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of the bars. The advantage claimed for this is the uniform distribution of the air over the entire surface of the grate and its thorough mixture with the gases of combustion. This type of grate is also made for a natural draft.

Loss Through Shaking Grates.—There is a certain amount of loss from unburned coal which falls through into the ash pit when

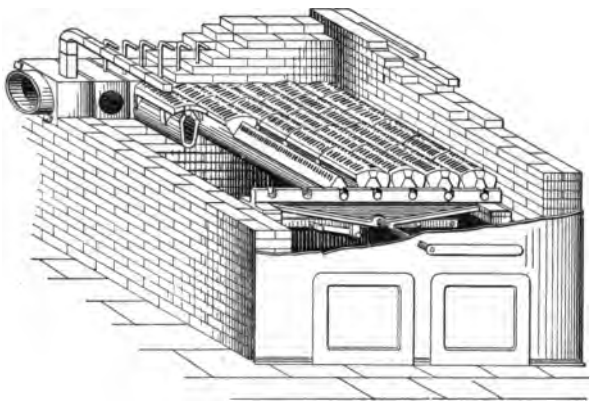

Fig. 38. White Hollow Blast Grate.

shaking grates are used. This usually amounts to from 1 to 3 per cent of the total coal consumed, even with the most careful handling, but is more than offset by the improved combustion due to the greater freedom from ash and clinker.

Mechanical Stokers

In large plants where it is desired to reduce the labor of firing, mechanical stokers are used. With a good form of stoker one man can care for several furnaces. Other advantages are more even firing, better combustion, and the introduction of coal with closed furnace doors. When properly designed these stokers feed both coal and air at a regular rate, and may be so adjusted that when the coal is burned in a suitably constructed furnace combustion of the gases will take place with the production of but little, if any, smoke. The chief objection to mechanical stokers is their first cost, but in plants of large size this is usually more than offset by increased efficiency and a reduction in the cost of firing.

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Over-fed Stokers.—Stokers are of two general types, known as "over fed" and "under fed." The Roney stoker, Fig. 39, illustrates one of the former. Fuel is fed into the hopper from which it is gradually forced by the pusher over the dead plate, onto the grates, which form a series of steps descending to the dumping grate near the bridge wall as shown. The grate bars are coupled

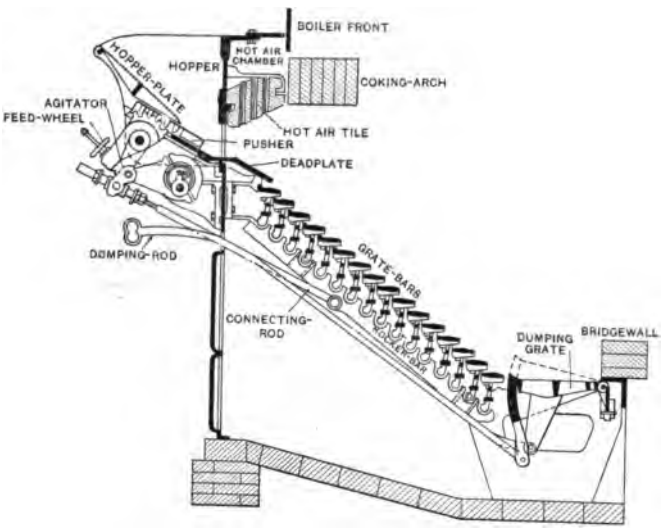

Fig. 39. Roney "Over-fed" Stoker.

together by a rocker bar which is given a backward-and-forward motion, thus causing the fuel to slide forward a short distance at each stroke. The rate of feeding is controlled by the length of the stroke of the pusher, or by changing the number of strokes per minute. All ashes and cinders are deposited on the dumping grate near the bridge wall. These stokers are usually operated by a small independent steam engine placed at one side of the boiler setting.

The Acme and Wilkinson stokers are somewhat similar to the one just described, both having the inclined grates with movable bars giving a forward movement to the coal which is fed into a hopper in front. The peculiar feature of the Wilkinson stoker is the use of hollow grate bars through which air is forced by means of small steam jets.

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Another form of the over-fed type is the Duluth stoker shown in Fig. 40. The grate bars in this case are carried by endless chains which run over sheaves at the rear of the furnace and sprockets in front, from which they receive their motion. The coal is fed into a hopper in front and is regulated by an adjustable distributing plate which extends across the grate at a height above it to give the desired thickness of fire. The stoker is mounted on trucks and can be easily drawn out of the furnace for inspection and repairs.

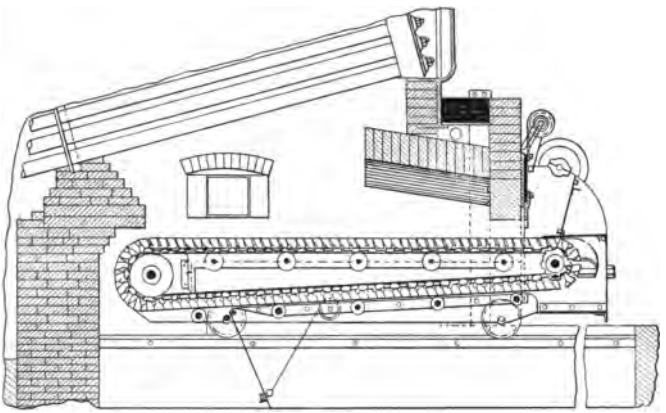

Fig. 40. Duluth Chain Grate Stoker.

Other chain-grate stokers are the Babcock and Wilcox, the Playford Chain Grate stoker, the Coxe Automatic stoker and the Green Traveling Link stoker. There are a number of other over-fed stokers of different forms but operating on the same general principles as those already described.

Under-fed Stokers.—The American stoker, Figs. 41 and 42, is of the under-fed type, and requires a forced draft for its operation. Just below the coal hopper, and connecting with it, is the conveyor pipe which in turn connects with the coal magazine. A screw conveyor or worm is located in the conveyor pipe and extends the entire length of the magazine. Air under pressure is brought into the space between the magazine and the outer casing and is discharged through openings in the air blocks around the upper edge of the magazine. This is shown more plainly in Fig.

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42, which represents a cross section through the furnace. Each stoker is driven independently by a small steam motor located in front and beneath the hopper.

The Taylor stoker, shown in Fig. 43, is an inclined under-fed stoker, in which the green fuel is delivered beneath the burning coal, while the entire fuel bed moves by gravity down an incline. In operation, coal from the hopper is fed into the retort, from which two cylindrical rams, assisted by gravity, introduce

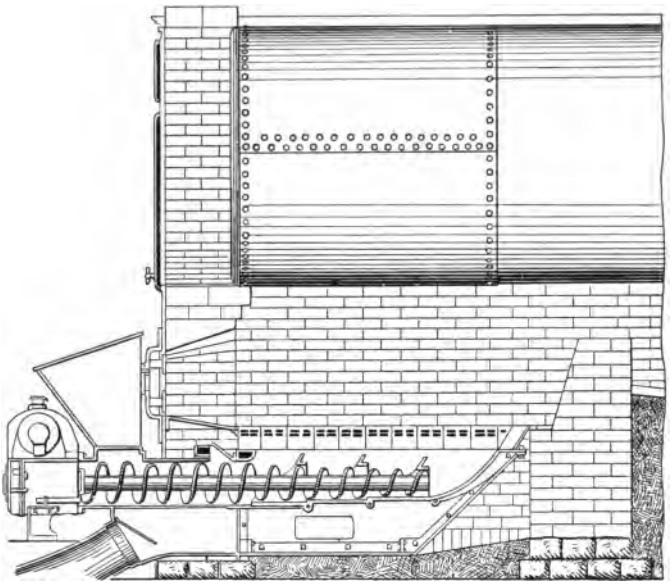

Fig. 41. American Under-fed Stoker.

it into the furnace at an angle to the fire surface. Movement of the upper ram pushes the green coal outward and upward, properly distributing it in the coking zone. The action of the lower ram is similar, but insead of bringing in fresh coal it pushes the fuel bed and refuse toward the dump plates at the rear.

The retort or fuel magazine is formed by two tuyere boxes; that is, the retorts and tuyere boxes alternate, the number depending upon the size of the boiler. A series of tuyeres is supported on each tuyere box, with openings in the vertical faces to distribute air under forced draft, to the fuel.

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Special Furnaces

The furnaces illustrated thus far have been of the usual form except for such slight changes as are necessary to adapt them to the various grates and stokers which have been described. We will now show two or three forms of special furnaces designed with particular reference to the complete combustion of the gases and the prevention of smoke.

Hawley Down-Draft Furnace.—In this furnace there are two

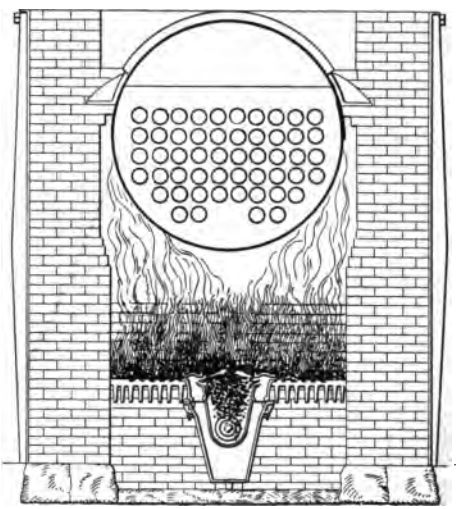

Fig. 42. Cross Section of American Stoker.

grates about a foot apart as in Fig. 44. The coal is fed upon the upper grate and as it becomes partly consumed, it falls through to the grate below where the combustion is completed. The draft is downward through the upper grate and upward through the lower, the gases being taken off from the space between them. The greater part of the air required for combustion is admitted through the fire door of the upper grate.

The upper grate in this furnace is made up of water tubes connected with the boiler, through which the water circulates. It is claimed that with this arrangement of grates the combustion is very complete and the amount of smoke produced correspondingly small.

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*The Wing Wall Furnace.**—This furnace as applied to a horizontal tubular boiler is shown in Fig. 45, and differs from the ordinary form of furnace by having two vertical walls, with a narrow opening between them, built back of the bridge wall near the end of the boiler. In addition to these, there is a fire-brick arch over the combustion chamber and a series of short walls or piers back of the bridge wall. Coal is fired alternately upon one

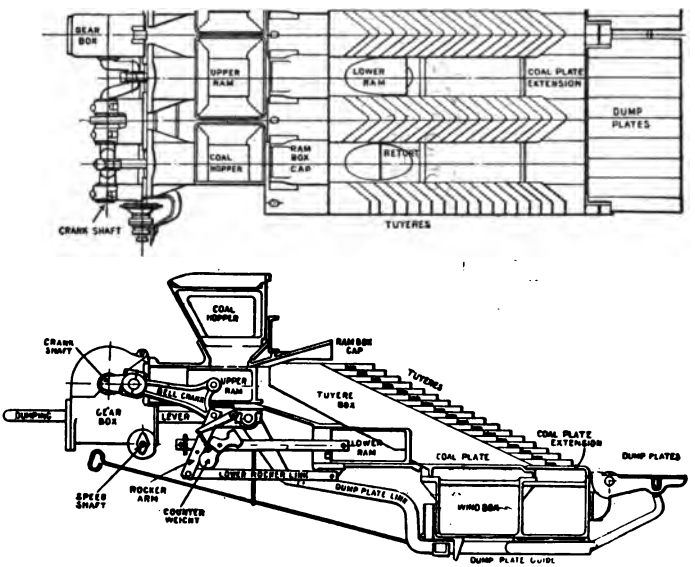

Fig. 43. Plan and Sectional Elevation of Taylor Stoker.

half of the grate at a time and the smoky gases from the freshly fired side and the excess of very hot air passing through the bed of bright coal on the other are brought together by passing through the narrow opening between the wing walls, thus making combustion much more complete. The fire-brick piers back of the bridge wall absorb a large amount of heat from the fire when it is hottest and give it out again to the cooler gases which pass into the combustion chamber after fresh coal has been fired.

The Kilgour furnace, shown in Fig. 46, consists of three essen-

*From *Steam Boilers*, by William Kent.

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tial parts, as follows: A hollow bridge wall (A), surmounted by cast-iron tuyeres; a transverse wall (B); and a combustion wall (C).

In operation the air supply is drawn from the ash pit by the effect of the draft and delivered through the tuyeres at the top of the bridge wall, where after reaching a high temperature, it mingles with the unburned gases from the fuel bed. After passing over the bridge wall, the mixture of gas and air is deflected

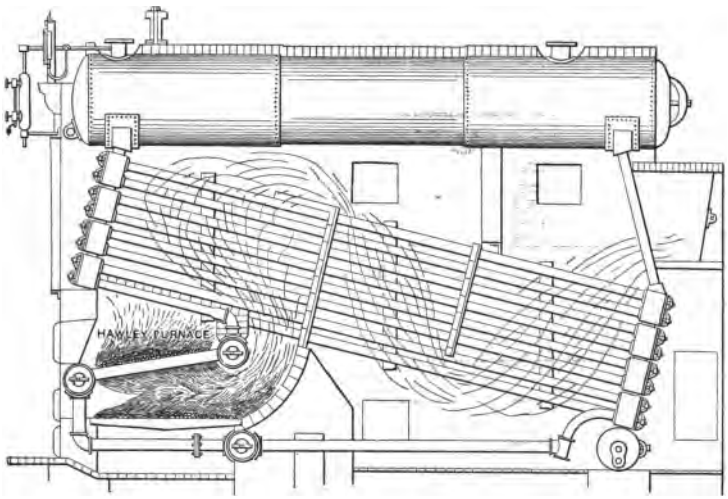

Fig. 44. Hawley Down-Draft Furnace.

downward by the transverse wall and enters the combustion tube where the mixing process is completed.

A small amount of water vapor is added to the distilled gases by means of a steam jet discharging into the hollow bridge wall below the tuyeres.

Dutch Oven.—A very efficient type of furnace for burning bituminous coal, is the Dutch oven, one form of which is shown in Fig. 47.

This furnace extends in front of the boiler and is fed through openings in the top. The mixing process in this case is performed by the fire-brick arch at the rear of the grate.

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Liquid Fuel

Method of Burning.—Oil fuel does not burn as a liquid, but must first be reduced to a vapor. After this change has taken place it is mixed with the right proportion of air and burned in a chamber kept at a proper temperature.

Means for vaporizing the oil, for insuring an adequate supply

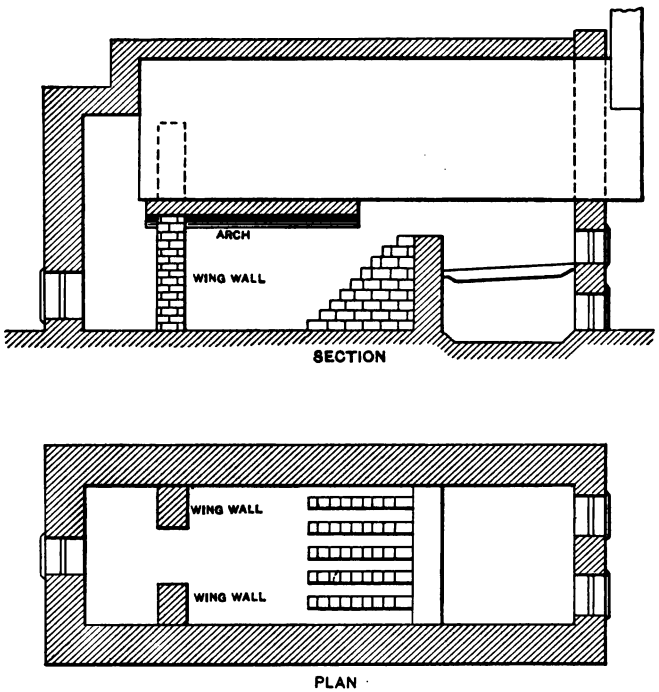

Fig. 45. Wing Wall Furnace.

of air, and for producing an intimate mixture within a chamber kept at a suitable temperature, must therefore form the foundation of all methods of oil burning.

Oil Burners.—There are many different forms of burners on the market, but they may be divided into three general classes, as follows:—

Mechanical spray burners, in which the oil is forced under high pressure through a nozzle made of such form as to break it up into a fine spray and thus render it inflammable.

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Air burners, where the liquid is held in suspension and driven into the furnace by means of a jet of compressed air.

Steam jet burners, in which the oil is injected in the form of a spray by means of a jet of steam, and the right amount of air allowed to mix with it, partly by suction through an opening in the burner and partly by way of an opening from the ash pit into the furnace.

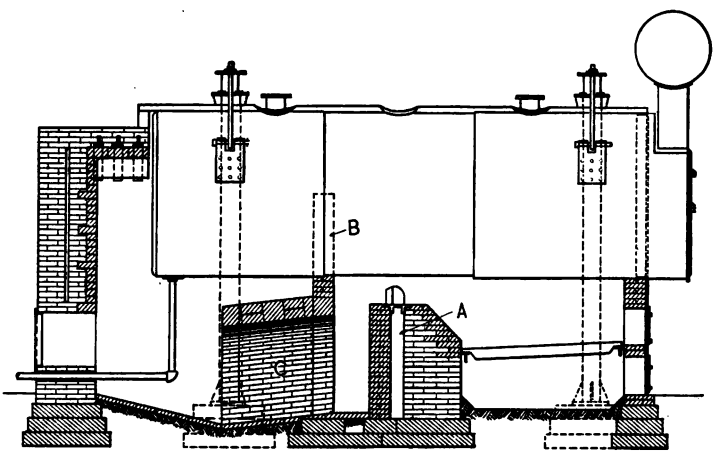

Fig. 46. The Kilgour Furnace.

Vapor burners are also used where the oil is first volatilized in a separate chamber and then burned in the furnace in the form of gas.

Description of Burners.—The Korting mechanical spray burner is shown in Fig. 48. The oil is first heated to a temperature of about 270° and then forced under a pressure of about 50 pounds per square inch through a nozzle having a small orifice in which is an adjustable spindle with spiral grooves. The rotary motion thus given to the oil causes it to break into a spray from the effect of centrifugal force as soon as it leaves the nozzle.

In another form the fuel passes through a minute orifice and is divided into a fine spray by striking against a cutter placed a short distance in front of the orifice.

The use of compressed air and steam in a burner is very much the same. In case air is used, it should first be heated to a high

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temperature by passing through a fire brick duct built into the furnace or combustion chamber, or by special means adapted to the particular type of burner in use.

The Grundell-Tucker air burner is shown in Fig. 49. Both the air and oil are supplied to the burner under a pressure of about

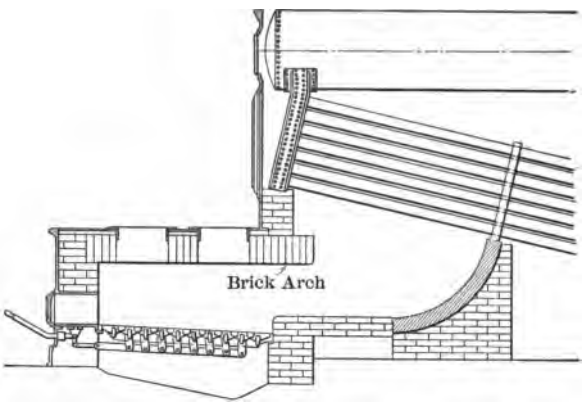

Fig. 47. Dutch Oven Furnace.

30 pounds. The oil is ordinarily heated by a steam coil to a temperature of 120 to 130 degrees, and the air to about 350 degrees. The high temperature of the latter is due in part to mechanical compression and partly to the absorption of heat while passing through an air chamber surrounding the burner.

Fig. 50 shows a section through the Hayes burner, which uses steam instead of compressed air. For securing the best results with burners of this type steam at a high pressure is desirable.

The Reed burner, shown in Fig. 51, is a combination steam and air burner. The air supply for burners of this design is usually heated by passing through flues built in the brickwork of the setting as already described.

In some other forms the oil is heated in the burner itself by forcing it through passages surrounded by steam.

Advantages of Different Types of Burners.—Tests made some time ago by the U. S. Government of different types of burners seemed to indicate that the best results were obtained by the use

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of either steam or air alone, rather than by a combination of the two within the burner. Burners using steam are considered preferable for large furnaces with much brickwork, as they spread the flame more and thus fill the entire space.

Air burners concentrate the flame at the point of contact, and are very destructive to plates and brickwork.

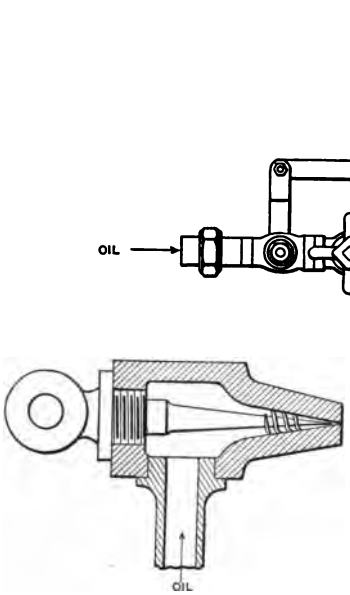

Fig. 48. Korting Mechanical Spray Burner.

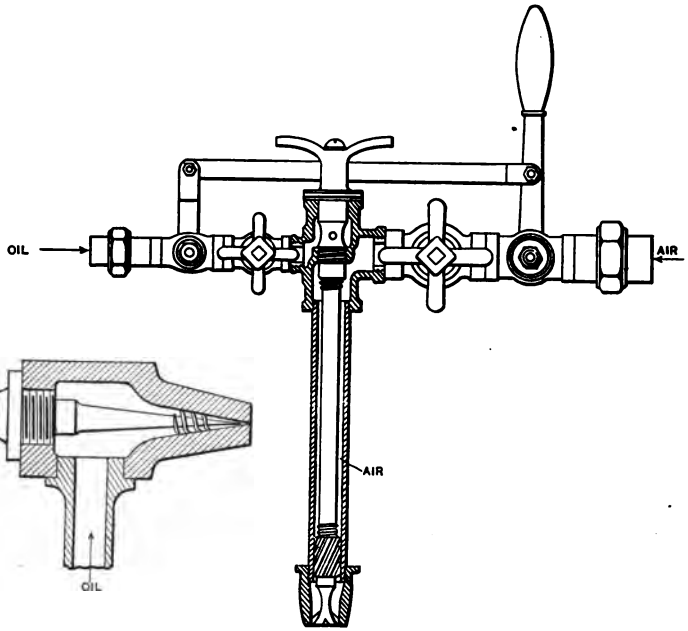

Fig. 49. Grundell-Tucker Air Burner.

Burners are made in all sizes up to 300 H. P. or more, but a number of smaller burners are found to use less fuel per horse power than a single large one. One burner for each 50 or 75 H. P. has been found to give the most satisfactory results under ordinary conditions. A burner placed in the center of each fire door is the arrangement usually employed.

Furnace Construction for Oil Fuel.—The construction of the furnace varies with the kind of boiler and type of burner used. The common method is to spray the oil against brickwork and along the grates, which should be covered with fire brick. This

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keeps the brickwork in an incandescent state and assists in the combustion. The covering for the grate may consist of broken fire brick laid in a loose bed so as to admit plenty of air. The grates should be of good length, with a bridge wall placed at the rear for the flames to strike against. Air may be admitted through the ash pit doors in the usual way. A form of furnace used with the Williams burner and others of the combined steam and air type is shown in Fig. 52. Another method is to cover

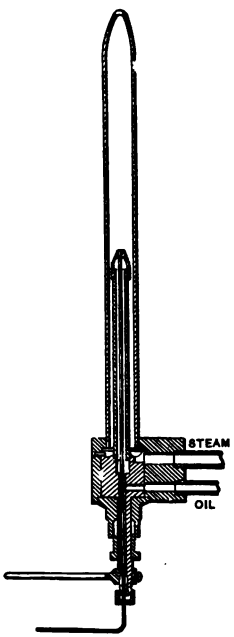

Fig. 50. Hayes Burner, which Uses Steam.

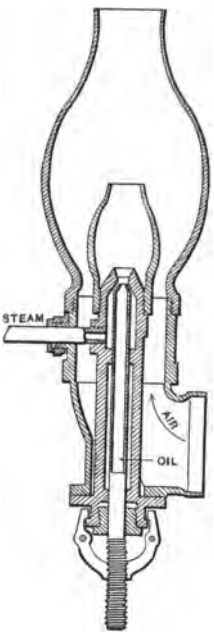

Fig. 51. Reed Combined Air and Steam Burner.

the grates with asbestos board, on top of which is loosely placed a layer of fire brick on the flat and the whole plastered over with fire clay. Uncovered spaces of about 70 square inches should be left under each 50 H. P. burner for admitting air from the ash pit. A "cascade" or "spatter wall" of fire brick on edge with narrow spaces between them is placed in front of the bridge wall, and the burners are so directed that the flame will strike the cascade wall about one third the way up from the bottom.

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Some types of burners can be bolted directly to the grate bars without change in the furnace construction.

Whatever kind of burner is used it should be so arranged that the flame will not be thrown directly against the boiler plates, for if this is done they will soon become overheated and badly injured. Manufacturers of burners will furnish detailed descriptions of the best form of furnace to be used in connection with their particular type of burner.

Storage of Oil.—The best way of storing oil for fuel is in an

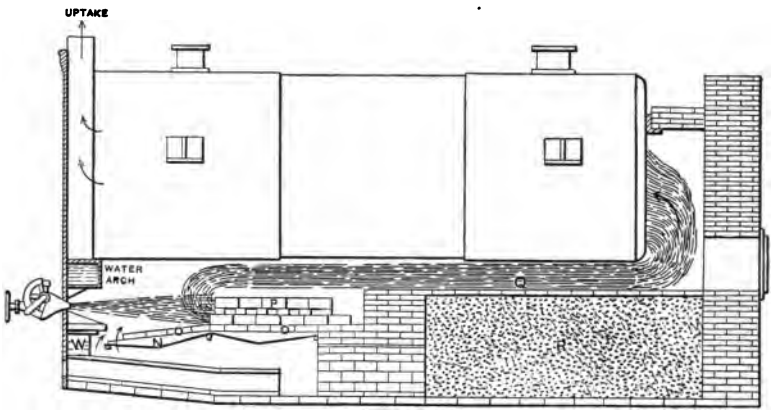

Fig. 52. Section of Furnace for Oil Burning.

underground tank made of steel plate and placed outside of the boiler room. The oil is either pumped to the burners or forced by means of steam or air pressure, admitted to the upper part of the storage tank. When steam is used for this purpose it also serves to warm the oil so it will flow more freely through the piping to the burners. If a pump is employed a steam coil is usually placed in the tank and supplied either with live steam or exhaust from the pump. In some systems the oil is warmed and purified by admitting hot water from the boilers into the lower part of the storage tank.

In order to get the best results, the oil should be supplied to the burner at a constant pressure. This is commonly done in one of two ways; a relief valve may be placed in the delivery pipe from the pump which will discharge back into the tank when the

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pressure reaches a given point, or the oil may be pumped to a standpipe, from which it flows under a constant head to the burners. When a standpipe is used the insurance companies require the use of an automatic valve which will drain the oil from the standpipe when the steam pressure drops below a given point.

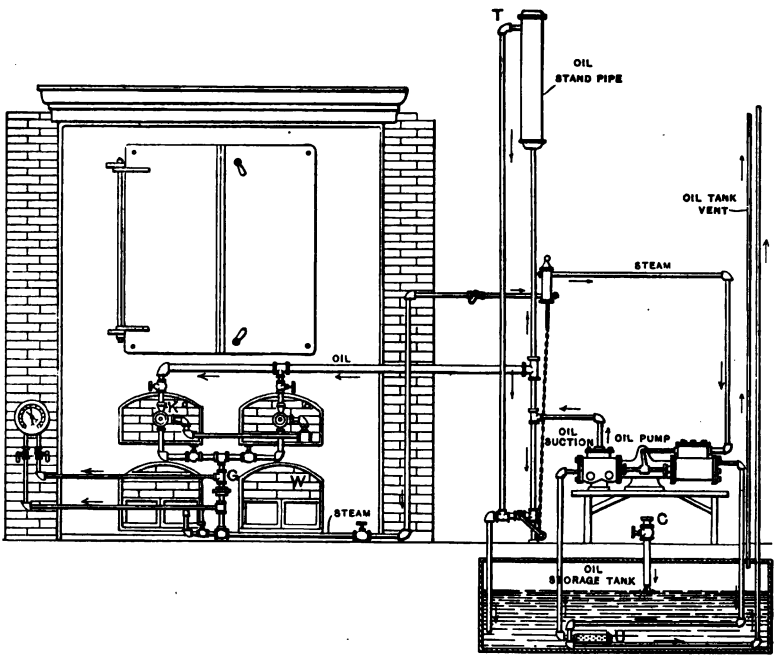

Fig. 53. Arrangement When Pump and Standpipe are Used.

This is to prevent flooding the furnace with oil when there is not sufficient pressure to atomize it. Fig. 53 shows the arrangement of pipe connections for using a pump and standpipe.

Danger from Liquid Fuel.—If the oil commonly used for fuel is heated much above 140° gases will be formed. These gases, while not explosive by themselves, will ignite when in contact with air and cause violent explosions.

Gases may be formed in three ways: First, by overheating, either from the sun or by nearness to the furnace; second, if air is forced through the oil it will become saturated with inflam-

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mable products; and third, gases will be formed if the oil is violently churned or agitated, as by passing it continuously through a pump by connecting the discharge from a relief valve directly with the pump suction. The oil itself is not easily inflammable, and a lighted match or red hot coal may be dropped into it without harm. With ordinary precautions the danger is very slight, and when used in connection with stationary boilers need not be seriously considered.

Some engineers recommended the use of a residuum from which the lighter products have been distilled to such a degree that the flash point is raised to a comparatively high temperature, thus practically eliminating the factor of danger under any conditions where the fuel is likely to be used.

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CHAPTER VIII.

BOILER SETTINGS

Foundations.—The first requirement of a good boiler setting is a substantial foundation. Where the soil is firm this may be either of concrete or rough stone laid in cement. The width of the foundation walls should be at least 16 inches greater than the walls of the setting which they support, thus allowing 8 inches upon each side. In the case of heavy settings for power work the foundation may be made from 20 to 24 inches wider than the brickwork, and in some cases a solid bed of concrete large enough to support the entire setting is provided. The depth of the foundation is usually made from 20 to 24 inches for boilers up to 48 inches in diameter and from 24 to 30 inches for larger sizes. These dimensions apply only where a solid earth footing is available, much heavier foundations being required if the earth is soft or yielding. Good concrete is obtained by using one part Portland cement, two parts sand, and four parts broken stone, the pieces of which should not be over 2 inches through. Concrete should be put down in layers not over 6 inches in thickness, each layer being thoroughly rammed before the next one is laid.

Foundations of this kind should be surrounded with board forms to hold the concrete while it is being laid. For the best results a Portland cement concrete foundation should be allowed to stand two months before loading. The time may be shortened somewhat, if necessary, provided the foundation is suitably proportioned and carefully watched, especially if the concrete is made as dry as possible.

Brickwork.—The setting above the foundation should be of hard-burned brick laid in freshly-made cement mortar made of one part Portland cement to two parts of clean, sharp sand. The furnace, bridge wall, and arching over the rear connection or combustion chamber should be lined with fire brick laid in pure cement or a mixture of fire clay and ground fire brick, with very close joints.

In setting boilers for power work it is customary to line the

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entire space exposed to the fire and hot gases with fire brick, and this is often done in the case of heating boilers in plants of considerable size. The outer walls, both at the sides and rear, should be made double with an air space between to reduce the conduction of heat through the walls.

Arch Bars.—One part of the setting which requires especial care is the construction of the arch over the back connection,

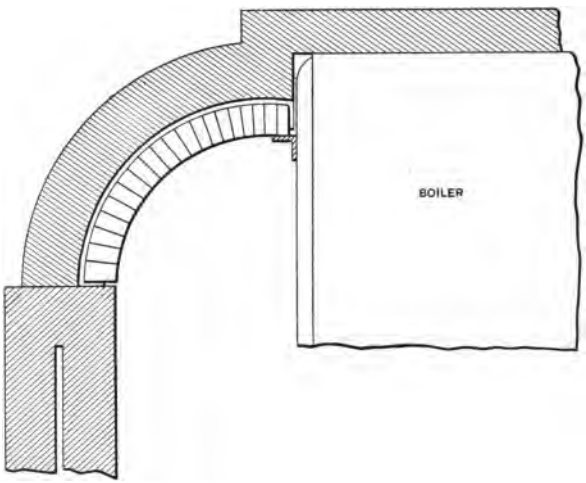

Fig. 54. Boiler Arch.

where the setting closes up against the back head of the boiler. This is sometimes turned from the back wall forward, as in Fig. 54. An angle iron is either riveted to the boiler head or supported in the side walls of the setting for the front of the arch to rest upon. The fire-brick lining is held in position by cast-iron arch bars and the whole construction covered over with brickwork. A space of at least an inch should be left between the arch and the end of the boiler to allow for expansion. One objection to this form of arch is the exposed position of the angle iron support, which is liable to be injured from contact with the hot gases which pass over it.

The effects of expansion and contraction of the boiler and setting are more apt to weaken the arch when made in this manner than when it is sprung from the side walls. In this case the arch is supported by bars so designed that a very limited part

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of the metal is exposed to the direct heat of the furnace gases. The form of bar used in the standard "Hartford Setting" is shown in Fig. 55. It consists of a curved piece of the dimensions shown, cast to a radius of 8 feet. The bricks are laid up against the bar in an arch as shown, and are held in position at either

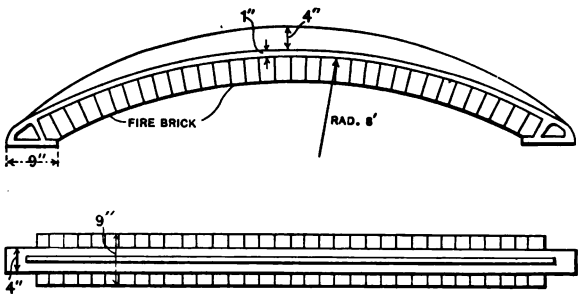

Fig. 55. Arch Bar.

end by projections cast upon the bar. In Fig. 56 the arches are shown in position. Three of these, arranged side by side, are in

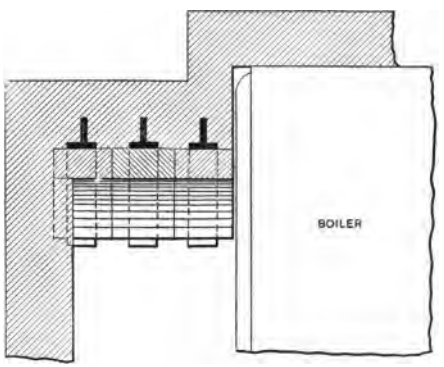

Fig. 56. Arch Sprung from Side Walls.

most cases sufficient to cover the space between the boiler and the rear wall. The ends of the iron supporting bars rest upon the side walls of the setting, and the rear wall is built up around the arch as shown.

Before placing the fire brick in the arch bar it should be turned bottom upward. If "arch brick" curved to an 8-foot

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radius can be had they should be used; if not, common fire brick will answer the purpose, but in this case every third or fourth brick should be beveled so that no large spaces will be left between them when they are in position. The last brick to be put in is

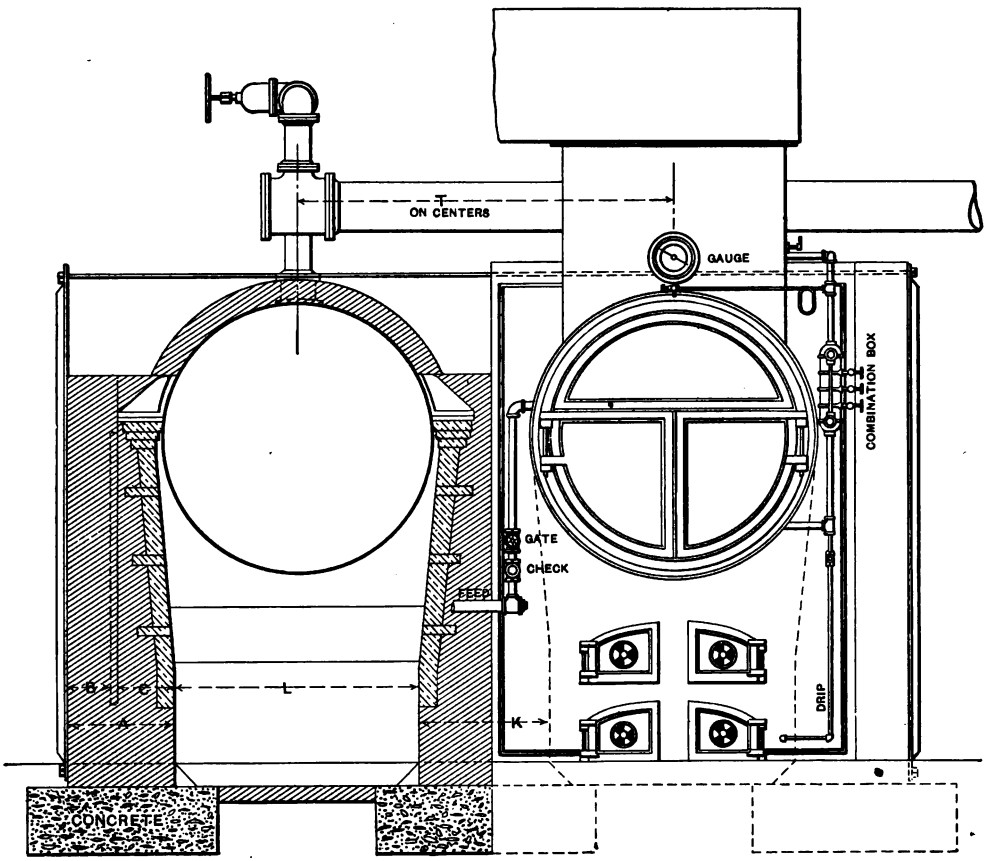

Fig. 57. Cross Section and Front Elevation of Boiler Setting.

the key brick in the center of the arch, which must necessarily be driven in endwise.

After the bricks have been carefully fitted in position, they should be removed, then wet and coated with a thin paste of fire clay and water, and put in place in the bar. The bar with the bricks in position, should next be turned right side up, and the

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TABLE XXVII.

DIMENSIONS FOR BOILER SETTINGS.

	Heavy setting for power.								Light setting for heating.							
Dia., inches.	30	36	42	48	54	60	66	72	30	36	42	48	54	60	66	72
A	22	26	26	26	30	30	30	30	22	22	22	22	22	22	26	26
B	8	12	12	12	12	12	12	12	8	8	8	8	8	8	8	8
C	12	12	12	12	16	16	16	16	12	12	12	12	12	12	16	16
D	7	7	7	7	11	11	11	11	7	7	7	7	7	7	11	11
E	8	8	8	12	12	12	12	12	8	8	8	8	8	8	8	8
F	12	12	12	12	12	12	12	12	8	8	8	12	12	12	12	12
G	8	8	8	12	12	12	12	12	8	8	8	12	12	12	12	12
H	22	22	22	26	26	26	26	26	18	18	18	22	22	22	22	22
I	24	24	24	28	28	28	28	28	20	20	20	20	24	24	24	24
J	8	8	8	8	9	9	10	10	8	8	8	8	9	9	10	10
K	24	28	28	28	32	32	32	32	24	26	26	26	28	28	28	28
L	24	30	36	42	48	54	60	66	24	30	36	42	48	54	60	66
M	20	22	22	24	24	24	24	24	20	22	22	24	24	24	24	21
N	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9
O	22	22	24	24	24	24	24	24	22	22	24	24	24	24	24	24
P	24	24	24	24	24	24	28	28	24	24	24	24	24	24	28	28
Q	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24
R	16	16	16	16	16	16	16	16	16	16	16	16	16	16	16	16
S	2-10	3- 4	3-10	4- 4	4-10	5- 4	5-10	6- 4	2-10	3- 4	3-10	4- 4	4-10	5- 4	5-10	6- 4
															7-10	8- 4
T	4- 0	4-10	5- 4	5-10	6- 8	7- 2	7- 8	8- 2	4- 0	4- 8	5- 2	5- 8	6- 4	6-10	7- 4	7-10
Length of setting = Length of tubes +	3- 6	3- 8	3- 8	4- 2	4- 2	4- 2	4- 2	4- 2	3- 2	3- 4	3- 4	3-10	3-10	3-10	3-10	3-10
Width of setting, 1 boiler	5- 8	6-10	7- 4	7-10	9- 0	9- 6	10- 0	10- 6	5- 8	6- 2	6- 8	7- 2	7- 8	8- 2	9-10	11- 4
															9- 4	9-10
Width of setting, 2 boilers	9- 8	11- 8	12- 8	13- 8	15- 8	16- 8	17- 8	18- 6	9- 8	10-10	11-10	12-10	14- 0	15- 0	17- 8	18- 8
															16- 8	17- 8
Width of setting, 3 boilers	13- 8	16- 6	18- 0	19- 6	23- 4	23-10	25- 4	26-10	13- 8	15- 6	17- 0	18- 6	20- 4	21-10	25- 6	26- 6
															24- 0	25- 0
Width of setting, 4 boilers	17- 8	22- 4	23- 4	25- 4	29- 0	31- 0	33- 0	35- 0	17- 8	20- 2	22- 2	24- 2	26- 8	28- 8	33- 4	34-10
															31- 4	32-10

NOTE.—Upper figures at bottom of last two columns to be used when width of grate equals diameter of boiler.

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spaces between them filled with a paste made of fire clay, ground fire brick and water. After the whole has dried and hardened, the arch may be place in position upon the setting.

Covering.—The top of the boiler is protected in some cases by springing a brick arch over it, leaving an air space of an inch or two between the brickwork and the shell, but more commonly light burned brick are placed loosely upon the side over the boiler shell, these in turn being covered with a layer of hard brick laid in cement mortar.

Iron Work.—From three to five pairs of buck stays with tie rods, depending upon the length of the boiler, should be provided to prevent the outer walls from spreading. Heavy cast iron plates about 20 inches in length, and planed upon the upper surface should be placed upon the side walls for the lugs or brackets to rest upon, and three iron rolls about 1 inch in diameter should be provided beneath each bracket, except the front ones, for carrying the boiler during expansion and construction. Special care should be taken to leave a clear space back of each movable bracket to allow freedom of expansion without bringing excessive strains upon the brickwork. The boiler front is made of cast iron, and provided with doors for access to the tubes, furnace, and ash pit.

Dimensions of Boiler Settings

Table XXVII. gives common dimensions for both light and heavy settings, and is to be used in connection with Figs. 57, 58, and 59.

The distance on centers between the boilers as given in the table is based on the assumption that the projection of the brackets from the boiler shell shall not exceed the figures given below.

Diameter of Boiler, inches	Maximum projection of brackets, inches
42	9
48	9
54	10
60	10
66	12
72	12

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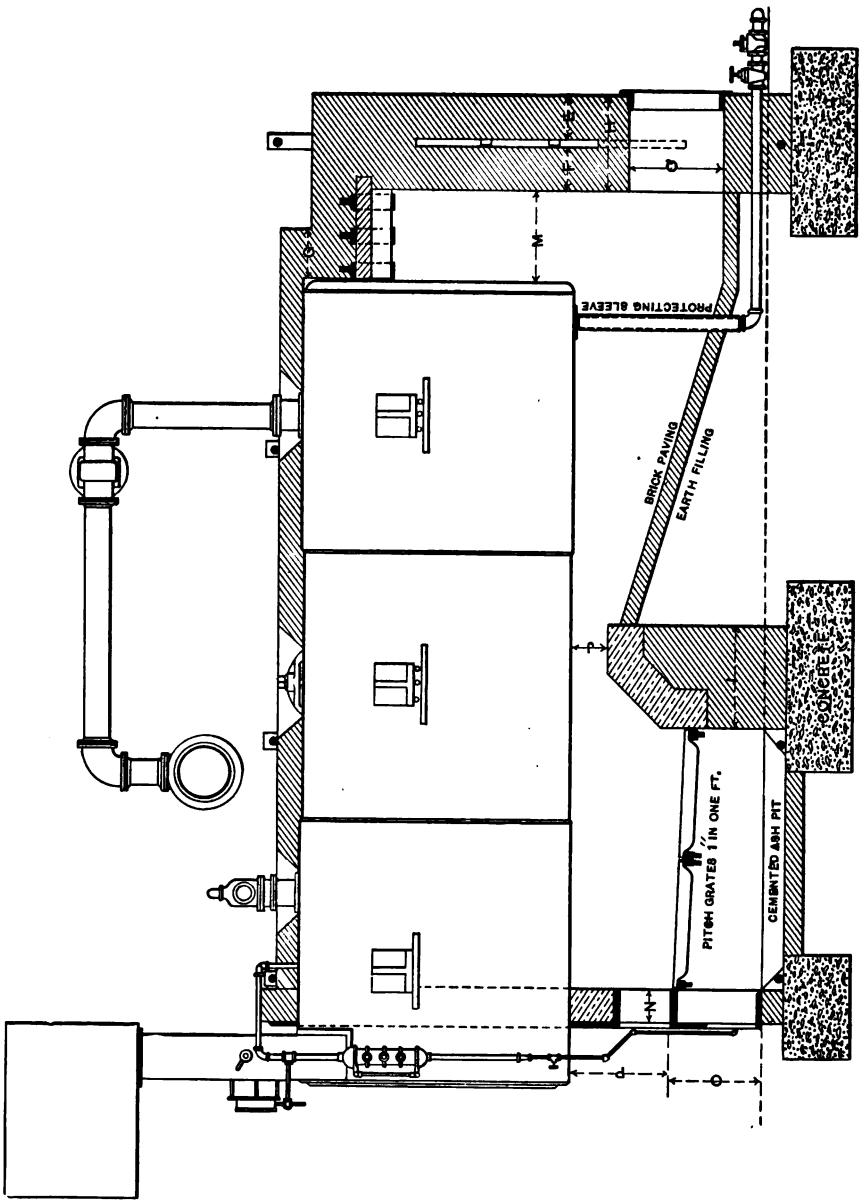

Fig. 58. Longitudinal Section of Boiler Setting.

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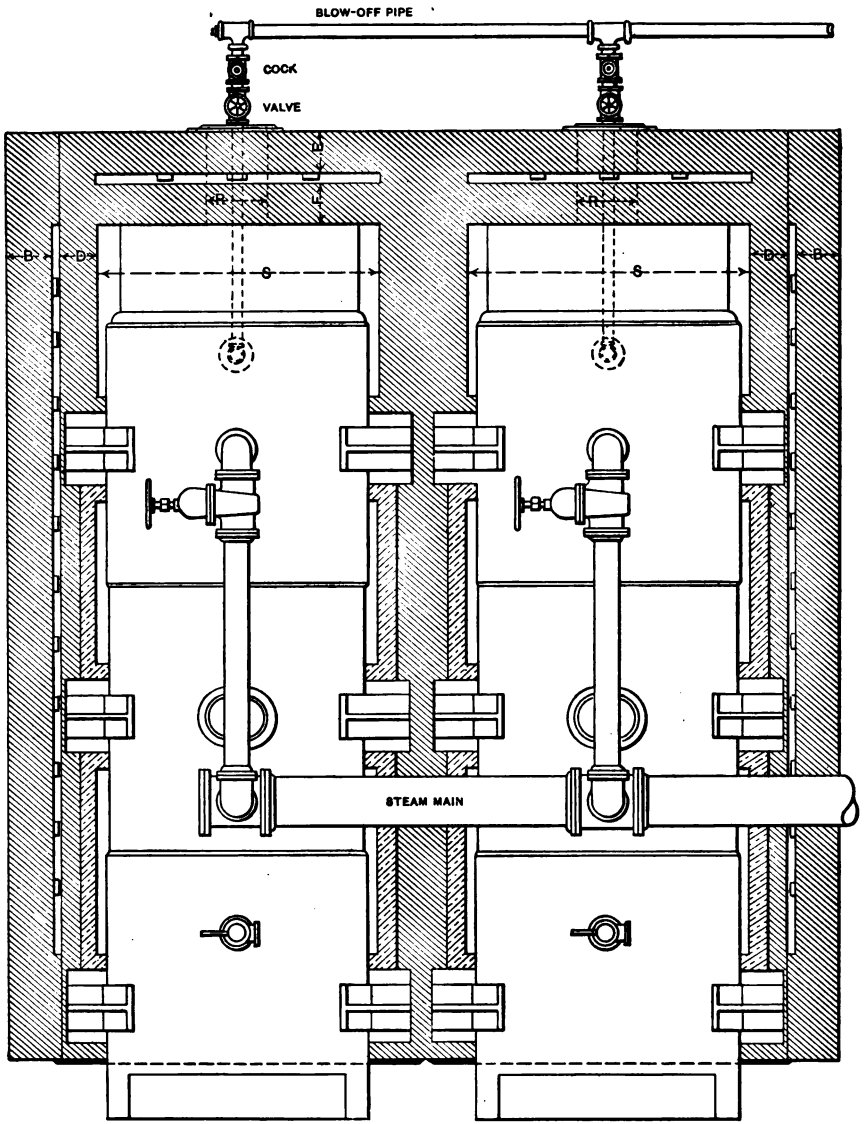

Fig. 59. Horizontal Section of Boiler Setting.

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If the projection is greater than given here, the thickness of the division wall must be increased accordingly.

Data for Design of Boiler Supports

Gallows-Frame Supports.—A form of support for tubular boilers which has come into quite common use is the metal

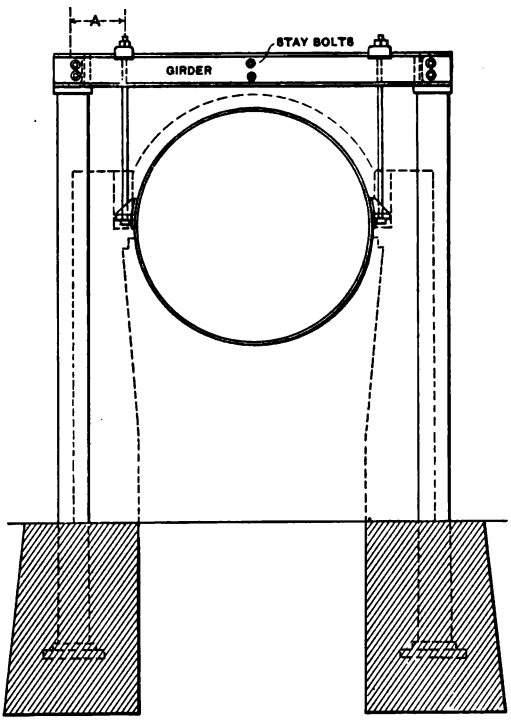

Fig. 60. Gallows Frame Support—End View.

gallows frame. Boilers supported in this manner are entirely independent of the brickwork, and when properly erected, expansion may take place without injury to the walls, which gives it a decided advantage over the common form of setting. There are several ways of constructing a support of this kind depending more or less upon the taste and skill of the designer. A very satisfactory arrangement, and one that is comparatively easy to

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construct, is shown in Figs. 60 and 61, which represent the end and side views respectively of a boiler supported in this manner.

The four columns are of wrought-iron pipe provided with flanges at the lower end and embedded from one fourth to one third of their length in concrete foundations.

The girders are made up of two channels placed back to back about 2 inches apart. A cast-iron cap having a tubular extension is placed on the top of each column, which should be trued in a

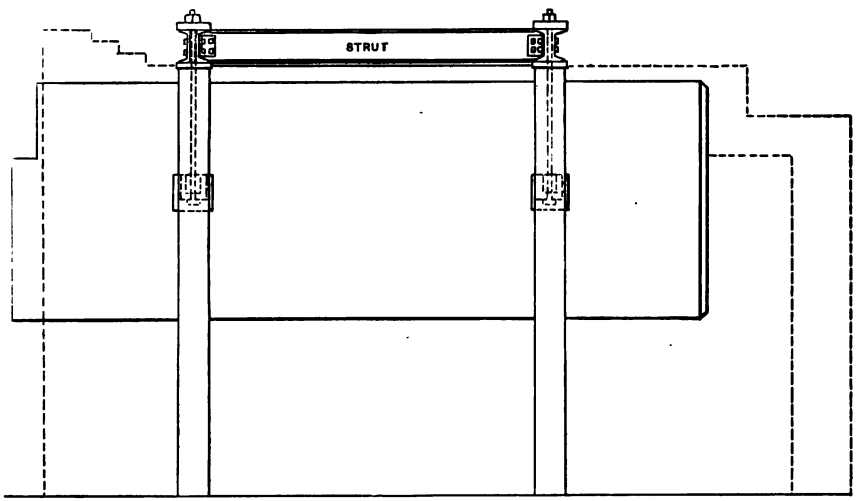

Fig. 61. Gallows Frame Support—Side View.

lathe, and is bolted in position. This cap has a flange, or stem, 2 inches thick extending to the top of the channel girders, which are bolted to it. The top or bearing surface should be beveled lengthwise of the girder from a point about an inch each side of the center as shown at *dd*, Fig. 62, in order that the load may always be applied at the center of the column. The suspension rods extend upward between the two channels and are provided with heavy nuts and washers at the ends. The washers distribute the load equally between the channels, and at the same time secure them against sidewise deflection by means of projections upon the under side, which are shown in Fig. 62. Two of these projections fit over the edges of the flanges while the remaining one fits tightly between the channel webs as shown.

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The channels may be still further stiffened by bolting them at the center as indicated in Fig. 60. The connection between the suspension rods and the boiler shell is made by means of a cast-iron lug as shown in Fig. 62. This is similar to the ordinary lug or bracket except the projection is made shorter to bring the rod as near to the shell as possible. Another method is to use wrought-

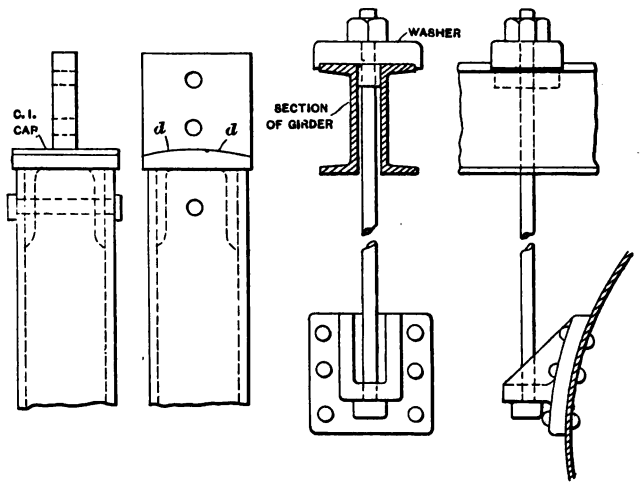

Fig. 62. Details of Gallows Frame Support.

iron hooks instead of lugs and attach them to loops at the ends of the suspension rods. In order to keep the front and rear girders the same distance apart and to still further strengthen the construction, it is well to put in two longitudinal struts, one over each side wall as shown in Fig. 61. Channels may be used for this purpose, fastened to the girders by means of angles and bolts.

When two boilers are set in a battery the gallows frame may be made as indicated in Fig. 63, or the support for the second boiler may a duplicate of that already described for a single boiler, the inner ends of the girder resting upon the same column which pases up through an opening in the division wall. The method shown in Fig. 63 is preferable on account of its simplicity, and the cost of the heavier girders and supports is usually less than that of the extra columns and foundations. The method of proportioning the various parts is as follows:—

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Suspension Rods.—The weight of the boiler full of water together with the piping which it supports, divided by 4, gives the load on a single rod, and this divided by 10,000 (the safe working stress per square inch) will give the required sectional area for a steel rod.

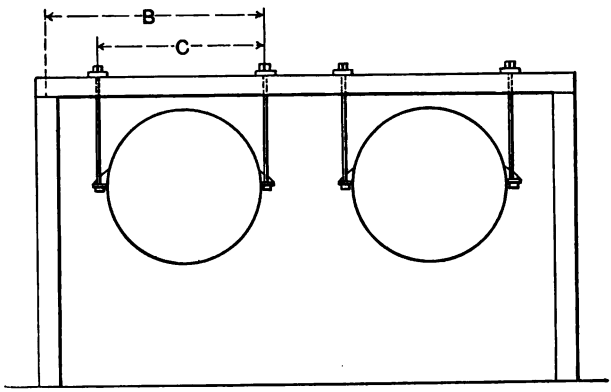

Fig. 63. Support for Two Boilers.

As the end of the rod has a thread cut on it, it should be upset to a larger diameter, in order that the sectional area at the bottom of the thread may be equal to, or greater, than that of the solid rod.

Table XXVIII. gives the necessary enlargement.

TABLE XXVIII.

DIMENSIONS OF SUSPENSION RODS.

Diameter of rod. Inch.	End upset to Inch.	Number of threads per inch.
1	1⅜	6
1⅛	1½	6
1¼	1⅝	5½
1⅜	1¾	5
1½	2	4½
1⅝	2⅛	4½
2	2¼	4½

Girders.—To determine the size of girder for a single boiler, multiply the load in pounds, on a single column (¼ the weight of boiler, etc.) by the distance in inches from the center of the

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column to the adjacent suspension rod. (See *A*, Fig. 60.) This gives the bending moment, which, in turn, divided by 10,000 pounds for steel, gives a quantity known as the section modulus. Table XXIX. gives the section modulus for different sizes of chan-

TABLE XXIX.

WEIGHT AND SECTION MODULUS OF CHANNELS.

Size of channel. Inch.	Weight per foot. Pounds.	Section Modulus.
6	8	4.3
7	9¾	6.0
8	11½	8.1
9	13¼	10.5
10	15	13.4
10	20	15.7
12	20½	21.4
12	25	24.0
12	30	26.9
15	33	41.7
15	35	42.7

nel bars, from which the required size can be selected, after determining the section modulus as above described. It should be borne in mind, however, that the load is equally divided between the two channels so that the section modulus of each need be only one half that computed for the whole girder.

For double settings (see Fig. 63) the load on each suspension rod is the same as for a single boiler but the load on each column and girder is twice as great. To find the bending moment in this case, multiply the load on a single column (¼ the weight of two boilers) by the distance *B*, Fig. 63; then multiply the load on a single suspension rod (¼ the weight of one boiler) by the distance *C*, and subtract the second product from the first; the remainder is the bending moment sought. Having determined the bending moment, the required size of the girder channels is found as before.

Columns.—The diameter of the pipe columns may be taken the same as the depth of the channels. This makes them amply strong for carrying the load and also gives a good appearance. Table XXX. has been computed by the methods just described for boilers of different sizes, for single and double settings. It will be noticed that for the larger sizes in the double setting,

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I-beams have been used in place of channels; this is on account of their greater strength for a given depth.

Masonry.—The walls in this type of setting may be lighter than

TABLE XXX.

DIMENSIONS OF GALLOWS FRAME SUPPORTS.

Diameter of boiler, inches	Length of tubes, inches	Diameter of suspension rods, inches	Diameter of threaded ends, inches	Single boiler.		Battery of two boilers.	
				Columns	Girders	Columns	Girders
48	11	1	1⅜	6-in. pipe	Two 6-in. channels, 8 lb.	10-in. pipe	Two 10-in. channels, 20 lb.
	13	1	1⅜	6 " "	" 6 " " 8 "	10 " "	" 10 " " 20 "
	15	1	1⅜	6 " "	" 6 " " 8 "	12 " "	" 12 " " 20¼ "
54	12	1	1⅜	6 " "	" 6 " " 8 "	12 " "	" 12 " " 25 "
	14	1¼	1½	7 " "	" 7 " " 9¾ "	12 " "	" 12 " " 25 "
	16	1¼	1½	7 " "	" 7 " " 9¾ "	12 " "	" 12 " " 25 "
60	13	1¼	1½	7 " "	" 7 " " 9¾ "	12 " "	" 15 " " 33 "
	15	1¼	1½	7 " "	" 7 " " 9¾ "	15 " "	" 15 " " 33 "
	17	1¼	1½	8 " "	" 8 " " 11¼ "	15 " "	" 15 " " 33 "
66	14	1¼	1½	8 " "	" 9 " " 11¼ "	15 " "	" 15 " " 33 "
	16	1¼	1½	8 " "	" 8 " " 11¼ "	15 " "	" 15 " " 35 "
	18	1½	2	9 " "	" 9 " " 13½ "	15 " "	" 15 " I-beams 42 "
72	16	1½	2	9 " "	" 9 " " 13½ "	15 " "	" 15 " " 42 "
	18	1½	2	9 " "	" 9 " " 13½ "	15 " "	" 15 " " 45 "
	20	1¾	2¼	10 " "	" 10 " " 15 "	15 " "	" 15 " " 55 "

in ordinary construction where they have to support the weight of the boiler. Two 8-inch walls, with a 2-inch air space between them will usually be sufficient. The foundations may also be shal-

TABLE XXXI.

DIMENSIONS OF PIERS FOR COLUMNS.

Diameter of boiler.	One boiler, Size of piers.	Two boilers, Size of piers.
In.	In.	In.
48	18x20	24x24
54	18x24	24x30
60	24x24	24x36
66	24x30	32x36
72	24x36	36x40

lower, from 12 to 16 inches being sufficient where there is a good earth footing. The piers supporting the columns should extend from 16 to 24 inches below the bottom of the columns. For a solid earth footing of well-packed gravel the dimensions of the piers may be as given in Table XXXI.

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Mr. George B. Francis gives the following permissible loads per square foot for different kinds of soil which will be found useful in designing the foundations for chimneys, boilers, engines, etc.: Hardpan, 8 tons; gravel, 5 tons; clean sand, 4 tons; dry clay, 3 tons; wet clay, 2 tons; loam, 1 ton.

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CHAPTER IX.

CHIMNEYS

The form and size of the chimney is an important factor in the design of a boiler plant, and should receive careful consideration.

Draft.—The draft of a chimney is due to the difference in weight of the hot gases inside of the flue and of the cooler air surrounding it. The outside air being heavier tends to fall, and flowing into the base of the stack through the boiler furnace pushes the lighter air upward, thus producing a constant current as long as a fire is maintained to heat the air inside of the chimney. The draft of a chimney is usually expressed in inches of water. This means that if one end of a U tube be connected with the chimney flue while the other end is left open to the atmosphere, the difference in level of the water in the two legs of the tube will show the intensity of the draft.

A difference of ¼ inch indicates a moderate draft, ⅜ to ½ inch a fairly good draft, and ¾ to 1 inch a strong draft.

Area and Height.—The required area of cross section for a given chimney depends upon its height and the amount and kind of fuel to be burned. The conditions which affect chimney draft are so numerous and difficult to treat theoretically that it is generally considered best to use empirical formulas, derived from the study of existing plants. The formulas most commonly used are those of Kent, Christie, and Gale, of which the first two are the simplest. The constant used in Kent's formula is based upon the assumption that 15 pounds of coal will be burned upon each square foot of grate surface per hour under the draft produced by a chimney 80 feet high. In this formula the effective area of the chimney is computed from the actual area under the assumption that the gases for a depth of 2 inches next to the lining have no velocity, due to friction. That is, the actual diameter of a round chimney, or the side of a square chimney should be 4 inches greater than the effective area required to

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carry off the gases. Kent's formula in its different forms is given below:—

$$E = \frac{.03\,H}{\sqrt{h}} = A - 0.6\sqrt{A}$$

$$H = 3.33\,E\,\sqrt{h}$$

$$S = 12\,\sqrt{E} + 4$$

$$h = \left(\frac{0.3\,H}{E}\right)^2$$

in which

E = effective area in square feet.

A = actual area in square feet.

H = horse power of boiler plant.

S = side of square in inches.

h = height of chimney in feet.

Table XXXII. has been computed by this method and will be

TABLE XXXII.*

SIZE OF CHIMNEYS WITH CORRESPONDING HORSE POWER OF BOILERS.

Dia. in inches.	Height of chimney in feet.											Effective area in square feet = E	Actual area in square feet = A	Side of square in inches of approximate area.
	50	60	70	80	90	100	110	125	150	175	200			
	Boiler horse power.													
18	23	25	27	.....	.....	.....	.....	.....	.....	.....	.....	.97	1.77	16
21	35	38	41	.....	.....	.....	.....	.....	.....	.....	.....	1.47	2.41	19
24	49	54	58	62	.....	.....	.....	.....	.....	.....	.....	2.08	3.14	22
27	65	72	78	83	.....	.....	.....	.....	.....	.....	.....	2.78	3.98	24
30	84	92	100	107	113	.....	.....	.....	.....	.....	.....	3.58	4.91	27
33	.....	115	125	133	141	.....	.....	.....	.....	.....	.....	4.47	5.94	30
36	.....	141	152	163	173	182	.....	.....	.....	.....	.....	5.47	7.07	32
39	.....	.....	183	196	208	219	.....	.....	.....	.....	.....	6.57	8.30	35
42	.....	.....	216	231	245	258	271	.....	.....	.....	.....	7.76	9.62	38
48	.....	.....	.....	311	330	348	365	389	.....	.....	.....	10.44	12.57	43
54	.....	.....	.....	363	427	449	472	503	551	.....	.....	13.51	15.90	48
60	.....	.....	.....	505	539	565	593	632	692	748	.....	16.98	19.64	54
66	.....	.....	.....	.....	658	694	728	776	849	918	981	20.83	23.76	59
72	.....	.....	.....	.....	792	835	876	934	1023	1105	1181	25.06	28.27	64
78	.....	.....	.....	.....	.....	995	1088	1107	1212	1310	1400	29.73	33.18	70
84	.....	.....	.....	.....	.....	1163	1214	1294	1418	1531	1637	34.76	38.48	75
90	.....	.....	.....	.....	.....	1344	1415	1496	1639	1770	1893	40.19	44.18	80
96	.....	.....	.....	.....	.....	1537	1616	1720	1876	2027	2167	46.01	50.27	86

* From Kent's Mechanical Engineers' Pocketbook.

found convenient for ready use. To find the pounds of coal that can be burned per hour with any chimney given in the table, multiply the corresponding horse power by 5. Only that

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part of the table is filled in which corresponds to ordinary proportions between area and heights, and these general relations should be followed if calculations are made from the formula.

In looking over the chimney plans of a number of important power plants, the heights were found to vary from 20 to 30 times the diameter of the flue, and a chimney designed with proportions within these limits should under ordinary conditions give a good efficiency, and at the same time present a pleasing appearance. In Christie's formula $H = 3.24\ A\ \sqrt{h}$, it being assumed that 4 pounds of coal are burned per H. P. hour. The notation in the above is the same as in Kent's formula.

Ratio of Grate to Chimney Area.—Another method which is sometimes used is based upon the relation between the grate and chimney areas. Common practice has shown that good results may be obtained if the area of the flue is made about 1/7 of the total area of all the grates connecting with it for chimneys 75 feet in height, and reducing it uniformly to 1/10 for heights of 200 feet. The required height of chimney to produce different rates of combustion under various conditions may be obtained by the following formula:—

$$\sqrt{h} = \frac{F \times W \times 0.1}{A}$$

in which

h = height of chimney in feet.

F = area of boiler grate surface in square feet.

W = pounds of coal burned per square foot of grate per hour.

A = internal area of chimney flue in square feet.

The square root of h for different heights is given in Table XXXIII.

TABLE XXXIII.

SQUARE ROOTS OF CHIMNEY HEIGHTS.

h	√h	h	√h
50	7.0	110	10.5
60	7.7	120	10.9
70	8.4	130	11.4
80	8.9	140	11.8
90	9.5	150	12.2
100	10.0	160	12.6

The formula will give approximately correct results for mod-

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erately free-burning bituminous, and medium to large anthracite coal; for free-burning bituminous coal deduct 10 per cent from height; for slow-burning bituminous add 10 per cent; for pea coal add 15 per cent, and for No. 3 buckwheat add 25 per cent.

The use of the above formula may best be illustrated by a practical example: What height of chimney will be necessary to burn 10 pounds of slow-burning bituminous coal per square foot of grate per hour under a battery of 6 boilers with a total grate area of 120 square feet? Area of chimney flue $= 120 \div 8 = 15$ sq. ft.

$$\sqrt{h} = \frac{120 \times 15 \times 0.1}{15}$$

$$\sqrt{h} = 12$$

Looking in Table XXXIII. we find that this corresponds to a height of chimney between 140 and 150 feet, which we will take as 145. Increasing this 10 per cent for the kind of coal used will give a required height of about 160 feet.

Height and Area for Different Coal.—Some engineers determine the height of chimney by the grade of coal to be used and then compute the required flue area by one of the methods already given. The following heights have been found to give good results in plants of moderate size, and produce sufficient draft to force the boilers from 20 to 30 per cent above their rating; Free-burning bituminous coal, 75 feet; anthracite of medium and large sizes, 100 feet; slow-burning bituminous, 120 feet; anthracite pea coal, 130 feet; anthracite buckwheat coal, 150 feet.

For plants of 700 or 800 horse power and over, the chimney should not be less than 150 feet regardless of the kind of coal to be used. It will be seen that for fairly high rates of combustion the formula gives a somewhat higher chimney, and for this reason is undoubtedly better adapted to power work when it is frequently desired to force the boilers or to change the grade of fuel. For heating purposes, or where the requirements are likely to remain more nearly constant, the latter method may be used with good results.

Form of Chimney.—It will be found for a given area that a round chimney will have a greater capacity and also present less resistance to the wind, but in ordinary building construction the

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flue is usually of rectangular form, especially in the case of smaller plants; although iron or steel stacks of circular section are often used. In the construction of brick chimneys it is better to make the inside walls entirely separate from the outside, both in order to permit free expansion of the inner wall and also to secure the advantages of the non-conducting effect of an air space between the inside and outside walls. Such a construction is common for the chimneys of power plants, but is not ordinarily applied to those in buildings and used for heating purposes only.

The external diameter or breadth of a chimney at the base should be at least one tenth the height unless it is supported by some other structure. The taper should be from ⅛ to ¼ inch per foot in height on each side.

Foundation.—The foundation of a large chimney is of great importance and should be laid out under the direction of an experienced engineer. The base of the foundation should be of such size that the pressure per square foot will not be over 1 ton for soft clay, nor over 2 tons for stiff clay, well-packed sand or loam. These loads are often exceeded in building foundations, but should not be in the case of chimneys.

When the soil is of low bearing power, pile foundations should be used, especially for large chimneys. The piles should be driven on 2 to 2½ foot centers and the tops cut off below the level of the surface water. A layer of concrete 2 or 3 feet in thickness should be laid over the piles with their tops extending into it, and on top of this the regular foundation of brick or stone.

In computing the weight of a chimney the following figures may be taken: Granite masonry weighs from 150 to 170 pounds per cubic foot, and brickwork from 120 to 130 pounds.

Thickness of Walls.—In designing the outer shell of a chimney the following approximate method may be used for determining the thickness of walls: If the inside diameter at the top is less than 3 feet the walls may be 4 inches in thickness for the first 10 feet, then increasing 4 inches for each 25 feet downward. If the inside diameter is more than 3 feet and less than 5 feet, begin with a wall 8 inches in thickness, increasing 4 inches for each 25 feet downward. If the diameter is over 5 feet, begin with a 12-inch wall, increasing downward as before. The lining or core

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may be 4 inches in thickness for the first 20 feet from the top, 8 inches for the next 30 feet, 12 inches for the next 40 feet, 16 inches for the next 50 feet and 20 inches for the next 50 feet. Using this method for an outer wall 200 feet in height and assuming a cubic foot of brickwork to weigh 130 pounds, it gives a maximum pressure of 8.2 tons per square foot of section at the base, while a lining 190 feet high would have a maximum pressure of 8.6 tons per square foot. The safe load for brickwork may be taken from 8 to 10 tons per square foot, although the strength of best pressed brick will run much higher.

When the thickness of walls is determined as above it is well to take sections of the chimney at different elevations and compute the weight of brickwork above, and see what the pressure per square foot of section is at these points, as a check. The lining of a chimney carries its own weight only, while the outer shell must be made stable against wind pressure also.

Wind Pressure.—Power to withstand the overturning force of the wind requires a certain relation between the height, breadth, weight, and form of chimney. This is expressed in the formula

$$\frac{c\,d\,h^2}{b} = W$$

in which

$c = 56$ for square chimneys.
$= 35$ for octagon chimneys.
$= 28$ for round chimneys.
$b =$ breadth or diameter at base, in feet.
$d =$ average breadth or diameter, in feet.
$h =$ height, in feet.
$W =$ weight of chimney, in pounds.

In checking a chimney for stability by the above, if the actual weight is found to be equal to, or greater than W given by the formula, then it will be stable against wind pressure.

Chimney Cap.—The external form of the top is a matter of appearance and may be designed to suit the taste. It should, however, be protected with a covering of tile or cast-iron plates to shed the rain and keep water from penetrating the brickwork. Iron ladders, either inside or outside the flue, and lightning rods are accessories which should be added to tall chimneys.

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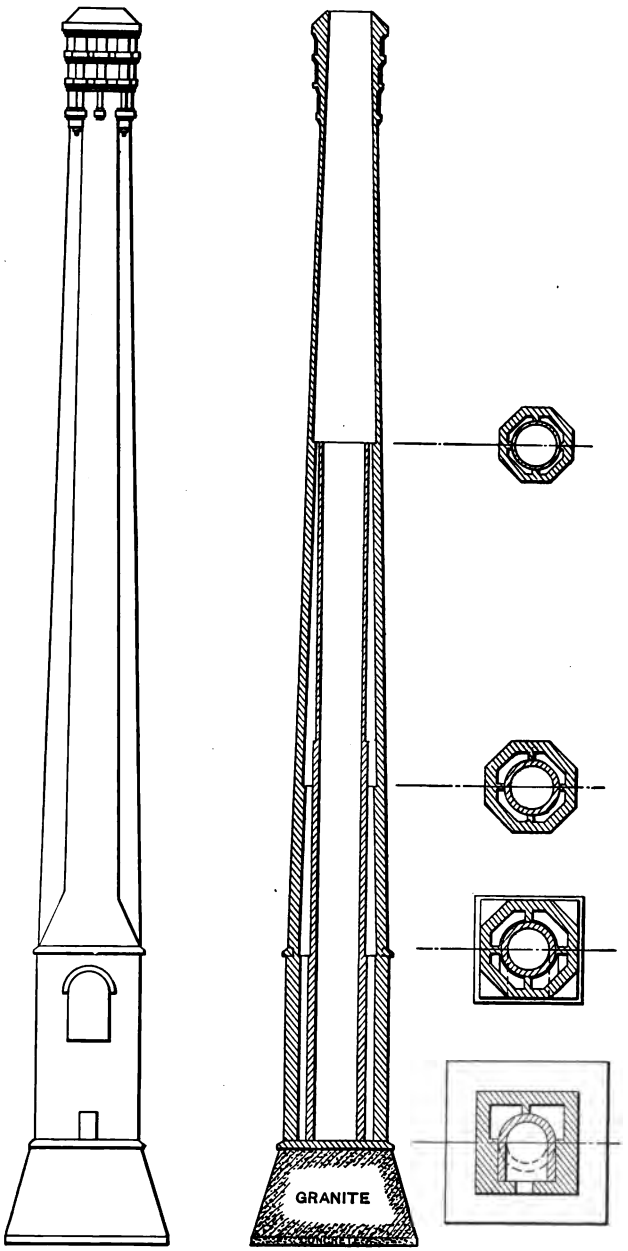

Fig. 64. Chimney Details.

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Lightning Conductors.—In the protection of a chimney against lightning it is essential that the points, if low, should be frequently spaced, and if a single point is used it should extend

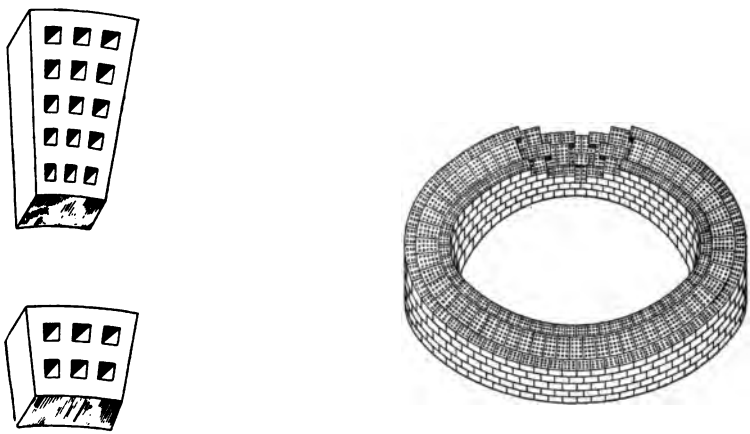

Fig. 65. Details of Alphons Custodis Radial Block Chimney.

above the top of the chimney a distance equal to 1.2 × the diameter of the flue. The points should be heavily gilded or coated with platinum, and should be in the form of cones

TABLE XXXIV.

DIMENSIONS OF STEEL STACKS.

Diameter of chimney in inches.	Height in feet.	Thickness of shell for first half of height in inches.	Thickness of shell for second half of height in inches.	Diameter of rivets in inches.	Pitch of rivets in inches.	Total weight of stack in pounds.
15	40	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{1}{2}$	3	1400
18	45	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{5}{8}$	3	2500
20	50	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{5}{8}$	3	3000
24	60	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{5}{8}$	3	4300
30	75	$\frac{3}{8}$	$\frac{5}{16}$	$\frac{3}{4}$	3	8300
36	90	$\frac{3}{8}$	$\frac{5}{16}$	$\frac{3}{4}$	3	11750
42	105	$\frac{3}{8}$	$\frac{5}{16}$	$\frac{3}{4}$	3	15700
48	120	$\frac{3}{8}$	$\frac{5}{16}$	$\frac{3}{4}$	3	21450

having a height equal to the radius of the base. They should be supported on brackets built into the shaft of the chimney, so that no bends or at the best but slight ones

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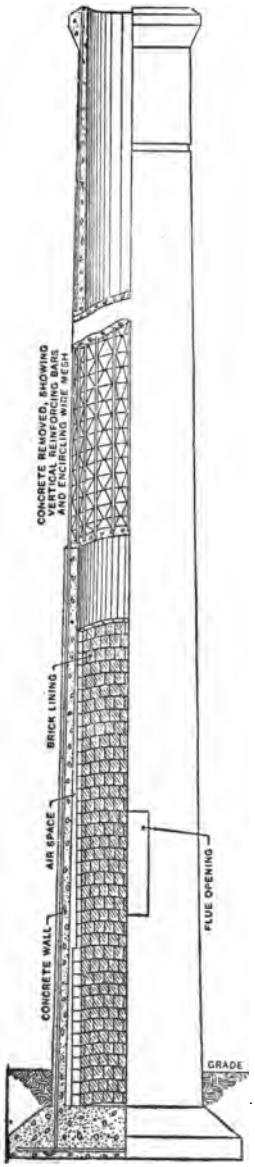

Fig. 66. Chimney Constructed of Reinforced Concrete.

will occur where the conductor clears the finial of the chimney.

Copper makes the best conductor, a tape or ribbon being used about ¾ inch wide by ⅛ to 3/16 inch thick. This should be carried down the chimney close to the ladder where it can be inspected from time to time. The ribbon should be secured to the shaft by copper clamps and no insulators should be used.

The ground plate should be of copper about 3 feet square and 3/16 inch thick and the ribbon should be brazed to plate. This plate should be buried in crushed coke in a moist location. If desired a sufficient length of the copper tape to make the exposed area 18 square feet can be buried in coke or the ribbon may be connected with iron water pipes. In dry localities extra precautions should be used. It is a good plan to locate ground plates where the water from the roof leaders can reach them at times of storm.

Lining.—The inner core or lining is usually built of fire brick for a portion of its height. The core may be from two thirds to three fourths of the total height of the chimney, although some engineers prefer to carry it up the full height. In any case it should be entirely separate from the outer wall.

Openings should be left near the base for smoke pipe and cleanout. These may be either arched over the top or provided with iron bearing

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bars. Fig. 64 will be found useful as a guide in the design of new chimneys.

Radial Block Chimneys.—Details of the Alphons Custodis radial block chimney are shown in Fig. 65.

Chimneys of this type are usually constructed without a core, as the hollow brick form a sufficient air space for insulating purposes.

When used in connection with chemical works, metal refineries, and furnaces where the temperature of the gases is such as to disintegrate the brick, a special lining is provided.

Reinforced Concrete Chimneys.—Chimneys constructed of reinforced concrete have come into quite general use within the past few years. These are of various designs, one of which is shown in Fig. 66.

Steel Stacks.—Stacks of boiler plate are quite commonly used at the present time. These are of two kinds; the first is designed to simply carry its own weight, and is supported against wind pressure by fastening to adjacent walls or by means of guy ropes. The second kind is made stable by heavy foundations and anchor bolts. Only the first will be taken up here. Steel stacks are made up in sections like the shell of a boiler, the end of one section being slipped into the next and the joint riveted.

Sometimes the sections are butted together and a covering ring placed over the joint and riveted to each piece. The stack usually rests upon a cast-iron ring or plate from 1 to 2 inches in thickness, which in turn is supported upon a foundation of stone or concrete.

When the side of the stack is cut away for connecting the smoke pipe, suitable means should be taken to strengthen the shell at this point. This may be done by riveting angle or tee bars to the plates at the sides of the opening, extending them for some distance above and below it.

Stiffening pieces are sometimes carried across the top and bottom of the opening, with a supporting piece placed vertically across its center. Table XXXIV. gives usual and safe dimensions for supported stacks.

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CHAPTER X.

MECHANICAL DRAFT

Mechanical draft is often used in large boiler plants, either as a substitute for a tall chimney or for increasing the power of a chimney already in use, and is also necessary with certain types of mechanical stokers.

Advantages.—With mechanical draft the boilers may be run at their maximum capacity without regard to outside weather conditions, and as the draft is independent of the temperature of the chimney gases it allows of their being cooled to a comparatively low degree, thus making available a certain quantity of heat which may be utilized in economizers for heating the feed water. A saving of 10 to 20 per cent in fuel is often made in this way.

Poorer grades of fuel may be used than is possible with a natural draft, and the capacity of a boiler may be increased from 30 to 40 per cent for continuous working, and much more for short runs. This is especially useful in the case of electric lighting and power stations, where the maximum quantity of steam is required for only a few hours per day. Forcing a boiler beyond a certain limit lowers its efficiency, but for short periods this is often more than offset by the reduced first cost of the plant and the smaller operating expenses.

Ease of regulation is another advantage of mechanical draft of considerable importance.

Rate of Combustion.—The rate of combustion with mechanical draft varies with the intensity of the draft, and in ordinary heating and power work usually ranges from about 15 to 30 pounds of coal per square foot of grate per hour. In marine and locomotive boilers it is much higher, varying from 30 to 50 pounds in the former, and from 60 to 125 in the latter.

Efficiency.—If the rate of combustion be increased in the case of a boiler running at maximum efficiency, while the grate area remains the same, the rate of evaporation per pound of coal will be reduced; but if the grate be made of such size that the *total*

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quantity of fuel burned per hour remains constant, a certain increase in efficiency will be observed.

This is due principally to the deeper fires which may be carried with a forced draft, and to the more intimate contact of the air with the coal. The increased pressure forces the air into spaces between the fuel which could not be reached under a lesser draft, and, as a greater proportion of the air is therefore consumed in combustion, the total quantity is reduced, resulting in a hotter fire and a greater radiation of heat to the boiler surfaces.

Grate Surface.—It has been stated in a previous chapter that under ordinary conditions a combustion of about 0.3 pound of coal per square foot of heating surface per hour has been found to give the most satisfactory results. This should still be kept in mind when applying mechanical draft to a boiler.

As the rate of combustion is increased the total amount of coal consumed should be kept approximately the same by reducing the size of grate. Assuming 15 square feet of heating surface per horse power for fire-tube boilers, and 10 for water-tube, we have the following grate areas per horse power for higher rates of combustion as given in Table XXXV.

TABLE XXXV.

GRATE AREAS PER HORSE POWER FOR MECHANICAL DRAFT.

Pounds of coal burned per sq. ft. of grate per hour.	Square feet of grate per horse power.	
	Fire tube boilers.	Water tube boilers.
15	0.30	0.22
18	0.25	0.19
21	0.22	0.16
24	0.20	0.14
27	0.17	0.12
30	0.15	0.11

Air Required for Combustion.—Theoretically, about 12 pounds of air are required to burn 1 pound of average coal or coke. In practice, however, with natural draft, it is necessary to supply about twice this amount to secure complete combustion, owing to the difficulty of bringing the air into contact with the entire body of coal. With mechanical draft and deeper fires, this quantity may usually be reduced to about 18 pounds per pound of coal, which is equivalent in round numbers to about 230 cubic feet, at a temperature of 60° F. As all of the air supplied has

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to be raised to the temperature of the furnace gases, this reduction should, theoretically, increase the efficiency of the boiler about 4 per cent under usual conditions. This is upon the assumption that the gases pass to the chimney at a temperature of 500° F. in each case. If an economizer is used, the gases may be cooled to a considerably lower temperature by the use of mechanical draft than is possible with a chimney, unless it be of excessive height, thus increasing the efficiency of the plant still more.

Intensity of Draft.—The draft, or pressure, required will vary in different cases, depending upon the quality and fineness of the fuel, the length and size of the smoke passages and the number of bends in the pipe. In ordinary practice with stationary boilers it usually runs from 1 to 2 inches of water column.

TABLE XXXVI.

AIR PRESSURE REQUIRED FOR DIFFERENT FUELS.

Kind of fuel	Pressure required in inches of water
Good steam coal,	0.4 to 0.7
Ordinary slack,	0.6 to 0.9
Very fine slack,	0.7 to 1.1
Semi-anthracite,	0.9 to 1.2
Anthracite slack,	1.3 to 1.8

Table XXXVI., tabulated by Dr. R. H. Thurston, gives the pressure required to properly burn different grades of fuel. If the smoke passages are long, or the flow of gases through them is restricted in any way, the pressures given in this table should be correspondingly increased.

Systems of Mechanical Draft

There are three systems of mechanical draft commonly applied to stationary boilers, as follows:—

1. The steam jet, by means of which steam and air are blown into the ash pit.

2. Forced draft or the closed ash pit system, in which a fan is employed to discharge air under pressure into the ash pit.

3. Induced draft, where an exhaust fan is placed in the smoke flue to produce a suction. In the latter arrangement the gases pass through the fan and are discharged into the chimney or uptake.

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Steam Jet System.—This is usually the most wasteful method of producing a draft, but has certain advantages in the burning of cheap coals high in ash, as the decomposition of the steam into hydrogen and oxygen in the bed of hot coal tends to prevent the formation of clinker. The expense of running a steam jet is that of the steam used and the heat required to superheat it to the temperature of the chimney gases. A series of tests made at the New York navy yard some years ago showed the amount of steam required to vary from 8 to 20 per cent of the total generated

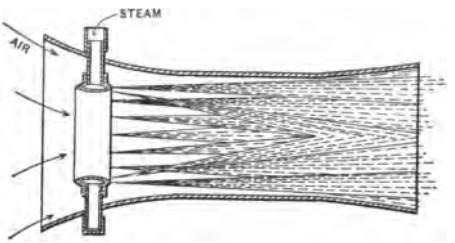

Fig. 67. Argand Blower.

by the boiler. The percentage required is found to be larger as the size of the plant diminishes. The simplicity and low cost of installation are among the chief advantages of the steam jet.

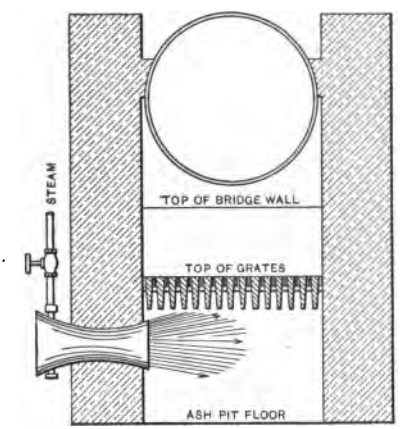

Fig. 68. Location of Argand Blower.

Fig. 67 shows the Argand blower, a form of steam jet commonly used with the McClave grate. It is connected with the ash

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pit as shown in Fig. 68, and delivers a large volume of air mixed with steam beneath the grates. The steam is discharged through a large number of small openings in a metal ring and the air is drawn into the surrounding nozzle by the strong suction produced. (See Fig. 67.)

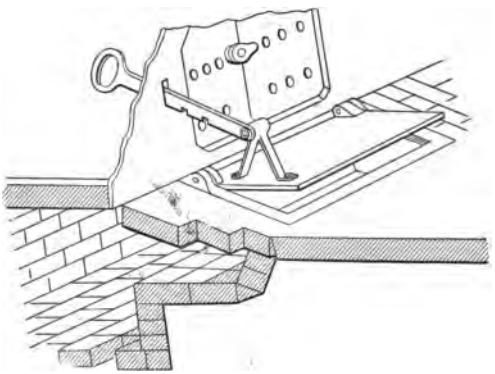

Fig. 69. Admission of Air for Forced Draft.

Closed Ash Pit System.—This system is the most easily applied to existing plants where it is desired to increase the draft

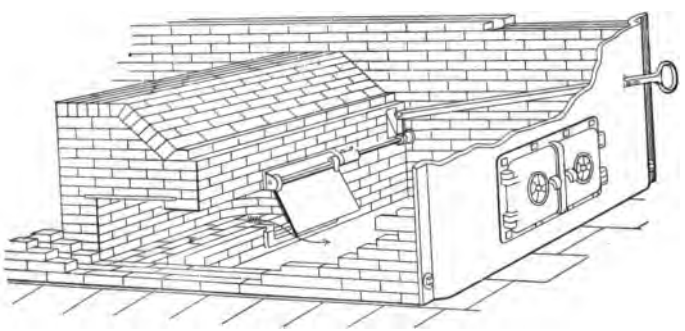

Fig. 70. Forced Draft—Damper in Bridge Wall.

for burning cheaper grades of fuel, and is also necessary in case under-fed stokers employing a retort are used. The air is usually introduced through openings in the bridge wall or in the bottom of the ash pit. Care should be taken to distribute the air as

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evenly as possible in order to prevent holes being blown through the fire. This may be accomplished by the use of deflecting dampers which throw the air downward as it enters the ash pit. The dampers used for this purpose by the B. F. Sturtevant Company are shown in Figs. 69 and 70. The pressure within the ash pit and furnace tends to blow ashes and flame into the fire room if forced too hard, but under a moderate draft but little

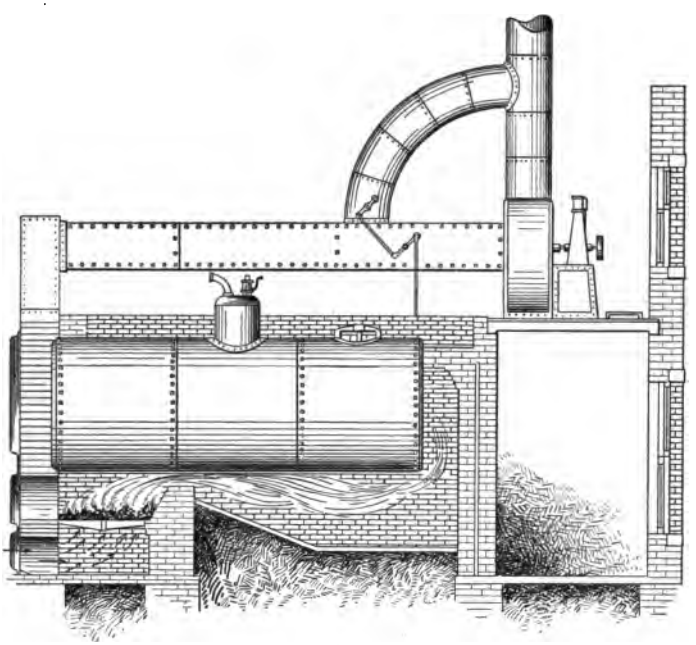

Fig. 71. Induced Draft System.

trouble will be experienced from this cause. Care should always be taken to close the damper in the blast pipe before opening the furnace or ash pit doors.

Induced Draft System.—With this system a partial vacuum is produced within the furnace, and its action is therefore practically the same as the natural draft of a chimney. It is more easily controlled than the other systems, and as the leakage is inward, all inconvenience from flame and dust at the fire and ash doors is avoided, and the fires may be made to burn more evenly over the entire surface than when forced draft is used.

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A specially constructed fan with self-cooling bearings is required for this system, as the gases passing through it are at a high temperature.

Fig. 71 shows in a general way the method of connecting the fan with the smoke pipe and chimney. This arrangement, however, can be varied to suit different conditions.

TABLE XXXVII.

FAN DIMENSIONS, QUANTITY OF AIR AND HORSE POWER FOR MAINTAINING PRESSURE IN ASH PIT.

For forced draft, temperature of air 60 degrees F.

Diameter of fan	Cubic feet of air delivered to furnace per minute	Pressure in inches of water.													
		.5		.75		1.00		1.25		1.50		2.00		2.50	
		Revolutions per minute	Horse power	Revolutions per minute	Horse power	Revolutions per minute	Horse power	Revolutions per minute	Horse power	Revolutions per minute	Horse power	Revolutions per minute	Horse power	Revolutions per minute	Horse power
Ft. In.															
2 6	4200	510	1.6	560	1.8	600	1.9	640	2.1	710	2.3	780	2.5	850	2.7
3	5800	430	2.2	460	2.4	490	2.6	530	2.8	590	3.1	640	3.4	710	3.8
3 6	7800	360	3.0	400	3.3	420	3.5	450	3.8	500	4.2	550	4.6	610	5.1
4	10000	320	3.9	350	4.2	370	4.4	400	4.9	440	5.4	480	5.9	530	6.5
4 6	12400	290	4.8	310	5.2	330	5.6	360	6.0	400	6.7	430	7.3	470	8.0
5	15200	250	5.9	270	6.4	290	6.8	310	7.4	350	8.2	380	8.9	420	9.8
5 6	18200	230	7.0	250	7.7	270	8.2	300	8.8	330	9.8	360	10.6	390	11.8
6	21400	210	8.3	230	9.1	250	9.6	260	10.4	290	11.5	320	12.5	350	13.9
7	28800	180	11.2	200	12.2	210	13.0	230	14.0	250	15.5	280	16.8	300	18.7
8	37200	160	14.4	170	15.7	190	16.7	200	18.1	220	20.1	240	21.8	270	22.5
9	46800	140	18.1	160	19.8	170	21.1	180	22.7	200	25.3	220	27.4	240	30.3
10	57400	130	22.2	140	24.3	150	25.8	160	27.9	180	3.1	200	33.6	210	37.2

Fans for Forced Draft

Both the "paddle-wheel" and "multi-vane" types of steel-plate fan are used for mechanical draft. The fan is usually driven by a direct-connected steam engine, although the belted type may be used if preferred. Regulation is generally obtained by means of a throttle valve in the steam pipe, which is operated automatically by the pressure in the boiler, thus changing the speed of fan as may be required to give the proper draft to maintain a constant boiler pressure.

Table XXXVII. gives size and speed of fan, quantity of air delivered, and horse power required for maintaining different pressures in the ash pit when the fan is located reasonably near

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the boiler. The air quantities are based on a velocity of 2,000 feet per minute through the fan outlet.

When the required volume of air comes between the quantities given in Table XXXVII. the speed may be increased or diminished to make up for small differences, but for any considerable variation the width of the fan should be changed, keeping the diameter and speed the same in order to maintain the pressure. which depends directly upon the peripheral velocity.

Fan Calculations.—The air volumes in the tables are computed on the basis that the width of fan wheel at the inlet is 0.52 of the diameter. This gives the widths tabulated in Table XXXVIII., from which the volume per inch in width can be computed for any given diameter of wheel.

TABLE XXXVIII.

FAN WIDTHS.

Diameter of wheel		Width of wheel
Ft.	In.	In.
2	6	16
3		19
3	6	22
4		25
4	6	28
5		31
5	6	34
6		37
7		43
8		50
9		56
10		62

Example.—It is desired to burn 1,800 pounds of coal per hour under a battery of boilers using a forced draft of 1.25 inches of water column. What size and speed of fan will be required, and what horse power of engine?

Pounds of coal per miuute $= 1{,}800 \div 60 = 30$, requiring $30 \times 230 = 6{,}900$ cubic feet of air.

From Table XXXVII. we find that a 3½-foot fan running at 450 revolutions per minute will deliver 7,800 cubic feet of air per minute under a pressure of 1.25 inches of water, and requires 3.8 H.P. The air volume required is 6,900 cubic feet, or $6{,}900 \div 7{,}800 = 0.88$ of that delivered by a standard wheel 22 inches in width, so that under the above conditions the required width of wheel would be $22 \times 0.88 = 20$ inches.

Theoretically, the horse power varies as the volume of air

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moved, but actually the friction of the fan forms a considerable part of the load, and this would be affected but little by any slight change in the width, so in practice it is best to take the horse power directly from the table without change.

In the above example a standard 3½-foot fan would probably be used without any reduction in width, and any excess of air would be cared for by automatic regulation of speed. In the case of large fans this is of more importance and must be dealt with accordingly.

Fans for Induced Draft

When induced draft is used, a much larger volume of air must be handled by the fan, due to the higher temperature and resulting expansion. The weight, and consequently the volume of air at 60° which is necessary for the combustion of a given weight of fuel, is of course the same whether forced into the ash pit or drawn in by suction, but in the latter case, when the heated gases reach the fan the volume is much increased, and the density correspondingly decreased.

If a fan supplying forced draft should be used to produce an induced draft of the same intensity the following changes would take place, their magnitude depending upon the temperature of the gases: First, the speed would have to be increased in order to maintain the same pressure; second, the power necessary to drive the fan at the higher speed would be greater than before; and third, the volume of air supplied to the furnace would be reduced.

Fan Calculations for Induced Draft.—In order to adapt Table XXXVII. to the conditions found in induced draft it is necessary to make certain corrections, which for the average flue gas temperature of 500° may be taken as follows: Volume $\times$ 0.73, speed $\times$ 1.35, horse power $\times$ 1.35.

Example.—A 4-foot fan when moving air at 60° F. will deliver to the ash pit of a boiler 10,000 cubic feet per miuute and maintain a pressure of 1 inch of water at a speed of 370 r. p. m., and requires 4.4 H. P. What will be the volume of air drawn into the ash pit by the same fan when used for induced draft, and what speed and horse power will be required to maintain the same pressure?

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Using the factors previously given, we have: Volume, 10,000 $\times$ 0.73 = 7,300 cubic feet; speed, 370 $\times$ 1.35 = 500 r. p. m.; horse power, 4.4 $\times$ 1.35 = 6.

Table XXXIX. has been computed in this manner and may be used for the conditions found in the average plant when it is desired to use the fan for induced draft.

TABLE XXXIX

FAN DIMENSIONS, QUANTITY OF AIR AND HORSE POWER FOR MAINTAINING PRESSURE IN ASH PIT.

For induced draft, temperature of flue gases 500 degrees F.

Diameter of fan	Cubic feet of air at 60 degrees F. drawn into furnace per minute	Pressure in inches of water													
		0.5		0.75		1.00		1.25		1.50		2.00		2.50	
		Revolutions per minute	Horse power	Revolutions per minute	Horse power	Revolutions per minute	Horse power	Revolutions per minute	Horse power	Revolutions per minute	Horse power	Revolutions per minute	Horse power	Revolutions per minute	Horse power
Ft. In.															
2 6	3000	688	2.2	756	2.4	810	2.6	864	2.8	958	3.1	1053	3.4	1147	3.6
3	4200	580	3.0	621	3.2	661	3.5	715	3.8	796	4.2	864	4.6	958	5.1
3 6	5700	486	4.0	540	4.5	567	4.7	607	5.1	675	5.7	742	6.2	823	6.9
4	7300	432	5.3	472	5.7	500	6.1	540	6.6	594	7.3	648	8.0	715	8.8
4 6	9300	390	6.5	418	7.0	445	7.5	486	8.1	540	9.0	580	9.8	634	10.8
5	11100	337	8.0	364	8.6	391	9.2	418	10.0	472	11.1	513	12.0	567	13.2
5 6	13300	310	9.5	337	10.4	364	11.1	405	11.9	445	13.2	486	14.3	526	15.9
6	15600	283	11.2	310	12.3	337	13.0	351	14.0	391	15.5	432	16.9	472	18.7
7	21000	243	15.1	270	16.5	283	17.5	310	18.9	337	20.9	378	22.6	405	25.2
8	27100	216	19.4	230	21.2	256	22.5	270	24.4	297	27.1	324	29.4	364	30.4
9	34200	189	24.4	216	26.7	230	28.5	243	30.6	270	34.1	297	37.0	324	40.9
10	41900	175	30.0	190	32.8	202	34.8	216	37.6	243	41.8	270	45.3	283	50.2

The use of Table XXXIX. is practically the same as that of Table XXXVII. The volumes in the second column are for an air supply at a temperature of 60°, computed in the same manner as for forced draft, while corrections have been made for volume, speed, and horse power.

Example.—Take the case of the previous example and apply induced draft instead of forced draft. The required air volume is 6,900 cubic feet per minute at a pressure of 1.25 inches of water.

Looking in Table XXXIX. we find a 4-foot fan will supply 7,300 cubic feet per minute, and will maintain the desired pressure at a speed of 540 r. p. m., and requires 6.6 H. P.

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Builders of mechanical draft apparatus usually furnish an engine of standard size with each fan, of sufficient power to drive it at the probable maximum speed under the conditions of each particular case.

Multi-Vane Fans. Tables XXXVII. and XXXIX. apply to the standard forms of "paddle-wheel" fans, in which certain general proportions are followed approximately by different manufacturers. When multi-vane fans are employed, special tables should be obtained from the makers, as this type of fan varies somewhat in construction and operation in its different forms, and general data cannot be used with satisfactory results.

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CHAPTER XI

BOILER ACCESSORIES

Water Glass and Gauge Cocks.—It is of much importance that the water level in a boiler be easily ascertained at any time. If it becomes too low there is danger of overheating the plates; and if too high, priming is likely to occur. The water level is indicated by the water glass and gauge cocks. The former consists of a strong glass tube about a foot in length, having the ends connected to the boiler by means of suitable fittings, so that the water level within it will be the same as that in the boiler, and always in sight. As the glass needs frequent cleaning or renewing, it is provided with valves for shutting it off from the boiler; and as it is liable to break occasionally from various causes, these valves are sometimes provided with small checks which close

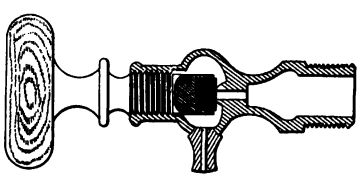

Fig. 72. Gauge Cock.

upon any sudden rush of steam or water through them. If dirty water is used or if foaming occurs, the level indicated by the glass is unreliable. If the steam and water connections are long they should be of extra large size.

There are many forms of gauge cocks in use, varying with different makers. The pattern shown in Fig. 72 is commonly used on boilers where they may be easily reached by the attendant. Others are especially adapted for use on water-tube or vertical boilers, where they are necessarily placed at some distance from the boiler room floor. These are usually opened by raising a weight or drawing out a valve stem, and may be operated by a chain or lever brought down to a point near the floor. There are usually three cocks, one placed at the highest point that it is

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desired to have the water rise, one at the lowest level, and one half way between. Instead of connecting the water glass and gauge cocks directly with the boiler head, it is more common to connect them with a cast iron chamber called a "combination box" or "water column," which in turn is connected at top and bottom of the boiler by 1¼-inch pipes. Fig. 73 shows a common arrangement. The botom of the glass is sometimes placed

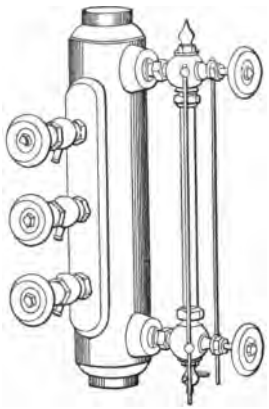

Fig. 73. Water Column.

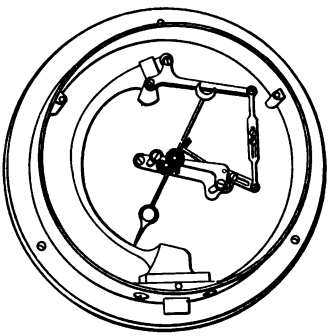

Fig. 74. Mechanism of Pressure Gauge.

on a level with the top of the upper row of tubes, but it is considered better practice to place it about 2 inches above the tubes,

TABLE XL.

SIZE OF FEED PIPE.

Diameter of boiler, inches	Size of feed, inches
36	1
42	1
48	1
54	1¼
60	1¼
66	1¼
72	1¼

so they will still be covered when the water disappears from the glass.

Feed Pipe.—The feed connections to a boiler, with the exception of that part exposed in the smoke bonnet, are always made of brass in the best class of work. The small section re-

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ferred to should be of extra heavy wrought iron. The branch to each boiler should be provided with a gate or globe valve and a check valve; the former being placed next to the boiler.

Table XL. gives suitable sizes of feed pipes for boilers of different diameters.

High and Low Water Alarms.—There are numerous devices for indicating the water level by means of floats so arranged that when the water becomes too high or dangerously low a whistle gives warning. Other forms are arranged to act both as a low

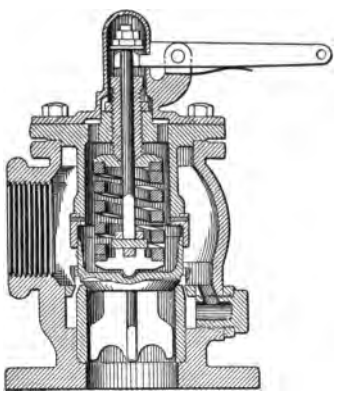

Fig. 75. Pop Safety Valve.

water alarm and automatic boiler feed. When the water level drops to a given point, the action of the float opens the feed valve and admits more water to the boiler. If for any reason the feed apparatus fails to work and the water level drops still lower, the whistle valve is opened and the alarm given.

Pressure Gauges.—The dial gauge is almost universally used. It is designed on the principle that a flattened tube, bent to a curve and closed at one end, tends to become straight when subjected to an internal pressure. The tube, which is usually oval in section, is bent into the arc of a circle as shown in Fig. 74. One end is fixed and connected with the boiler pressure. The other end is closed and free to move. By means of levers and a curved rack and pinion, the motion of the free end is multiplied and the internal pressure is indicated by a pointer upon a dial.

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The pipe connecting the gauge with the boiler should contain a bend or coil called a siphon, which will remain filled with water, thus preventing the steam itself from entering the tube. Whenever a siphon is selected care should be taken that it has a water capacity sufficient to fill the tube of the gauge. If the connections are such that the pipe leading from the boiler extends downward to the gauge, thus forming a pocket to become filled with water, a correction should be made by moving the pointer on its pinion to offset the weight of the water column which acts

TABLE XLI.*

DIAMETERS OF LEVER SAFETY VALVES.

	Lever Safety Valves.						
Diameter of valve, inches,	1	1½	2	2½	3	3½	4
Horse power of boiler,	20	30	40	50	60	70	80

* From Kent's Mechanical Engineer's Pocketbook.

TABLE XLII.*

DIAMETERS OF POP SAFETY VALVES.

Diameter of valve, inches	Horse power of boiler	Diameter of valve, inches	Horse power of boiler
1	5 to 10	3	75 to 100
1½	10 to 20	3½	100 to 125
1½	20 to 30	4	125 to 150
2	30 to 40	5	150 to 200
2½	40 to 75		

* From Kent's Mechanical Engineer's Pocketbook.

upon the gauge, in addition to the steam pressure. If a gauge is to be connected up in this way, the height of the water column should first be determined, and the necessary changes made before the gauge leaves the works. In case several boilers are set in battery, each should have its own pressure gauge, which should be connected directly to the boiler and not to the steam pipe.

Recording Gauges are often used in large plants for keeping a continuous record of the boiler pressure during the entire day. These are similar in construction to an ordinary pressure gauge, with the addition of a revolving paper dial run by clockwork. The pointer extends over the dial from one side, and the end is provided with a pen, which traces a line on the dial at a distance from the center proportional to the steam pressure. A glance at

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the record thus made will show the steam pressure at any time during the day or night.

Safety Valves.—These are constructed in several forms, but in every case they consist of a valve opening outward and held in place by a weight or spring. The lever safety-valve is the older form, and is used to some extent at the present time on low pressure heating boilers. The valve in this case is held in place by a weight suspended on a lever. The force required to lift the valve is regulated by shifting the position of the weight upon the arm. When the valve becomes leaky it should be made tight by regrinding; additional weights should never be added for this purpose. This type of valve has several defects, among which are the following: It does not close promptly when the pressure is reduced, it is likely to leak after it is closed, and it can be easily overloaded or wedged in place. The spring or pop safety valve, shown in Fig. 75, overcomes these defects in a large degree. These valves open when the steam pressure is sufficient to overcome the tension of the spring, and can be adjusted for different pressures by means of a nut and screw at the top, with increases or diminishes the tension as required. They are often provided with a "lock-up" attachment so that the regulating parts cannot be tampered with either by accident or design. The safety valve should be connected by a separate nozzle directly to the boiler without pipe or elbow, and should be opened every day by means of a lever and chain to see that it is in working order.

There are many rules for determining the size of a safety valve, each giving a different result. For lever valves the sizes given in Table XLI. may be used with satisfactory results. Where a diameter of over 4 inches is called for it is customary to use two smaller valves having an equivalent capacity.

For pop valves the sizes given in Table XLII. may be used. These correspond well with the average capacities given by different makers and with results obtained from the most approved methods of calculation.

Fusible Plugs.—Danger from overheating the plates of a boiler is guarded against by means of a fusible or safety plug, which consists of a brass bushing with a core of some fusible metal

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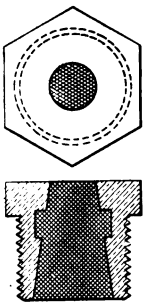

Fig. 76.
Fusible Plug.

which melts before an excessive temperature is reached. The government requires that pure Banca tin be used for this purpose on the boilers of steam vessels. In case of low water the plug melts and the steam and water blowing into the furnace will check the fire to some extent, and also give warning to those in charge. In a horizontal tubular boiler the fusible plug is screwed into the rear head at a point about 2 inches above the upper row of tubes; in a vertical boiler it is placed in the lower tube sheet or a little above the crown sheet in one of the tubes; and in the case of a water-tube boiler it is placed in the lower portion of the upper drum. To be efficient the plug should be of good length and the core made tapering in form, with its larger end toward the steam pressure to prevent its blowing out. Fig. 76 shows a common form to be screwed into the plate from the outside.

Injectors

Injectors.—The injector is used extensively in both stationary and marine practice and exclusively upon locomotives. Its advantages are its compactness and simplicity, having no moving parts when in operation except a light check in certain forms. In stationary work the injector is often used in connection with a pump as a safeguard in case of a breakdown. The successful operation of an injector depends upon the steam pressure, height of lift and temperature of the feed water. Hence, a proper combination of these conditions must be obtained, which limits its field of usefulness to some extent. The common form of injector combines two distinct functions, the *lifting* and the *forcing* of the water.

Principle of Operation.—Fig. 77 shows in diagram the essential parts of the ordinary injector. Steam is admitted through the nozzle E and passes through a second nozzle F. The friction between the rapidly moving jet of steam and the air lying between it and the sides of the nozzle F causes some of the air to be dragged along, thus producing a reduction of pressure in the

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chamber *A*. A partial vacuum being formed in the suction pipe *B*, water is drawn into the injector and thus the first part of its operation is accomplished. The second part, that is, the forcing

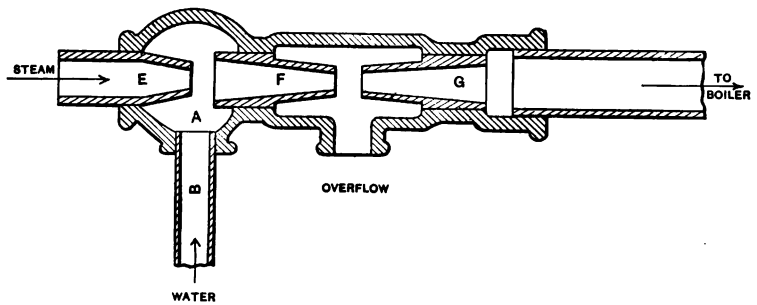

Fig. 77. Diagram of an Injector.

action may be explained as follows. The jet of steam issuing from the steam nozzle *E* is condensed by the surrounding water and much reduced in size, while its momentum remains the same.

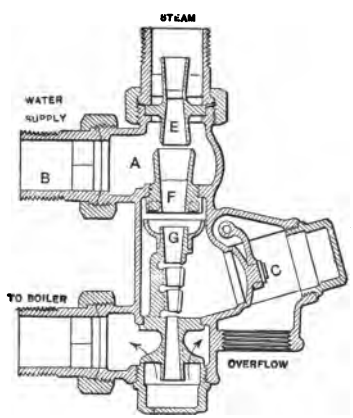

Fig. 78. Lunkenheimer Injector.

This being the fact it is able to enter a much smaller opening than that from which it issued, thereby correspondingly increasing the pressure per square inch of section required to stop it. This increase in pressure is so great that the jet not only has sufficient energy to reënter the boiler against its own pressure, but has

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enough surplus energy to carry in with it a certain amount of feed water.

Referring again to Fig. 77, the steam enters the nozzle *E* and flows through the combining or mixing tube *F*, and is discharged through the overflow. This action soon exhausts the air from

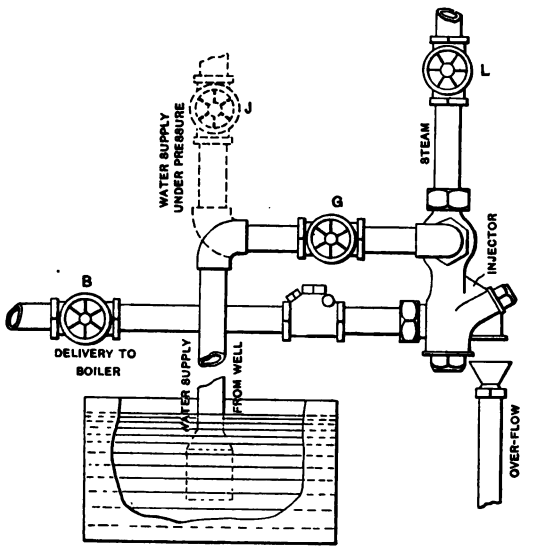

Fig. 79. Connections for Lunkenheimer Injector.

the interior of the injector, as already described, and water is drawn up through the suction pipe and discharged through the overflow with the condensed steam. As soon as the water supply has been cut down to the right amount the strength of the jet becomes sufficient to force its way into the boiler through the discharge or delivery pipe *G*. If the injector were made as shown in Fig. 77 there would be a constant dripping from the overflow, because the jet would never be of exactly the right size to enter the tube *G* without spattering more or less water from the edges of the opening. This is avoided in practice by making the overflow opening at the top and providing it with a light check valve which prevents the dripping of water when in use, and yet offers but little resistance when first starting the injector. In-

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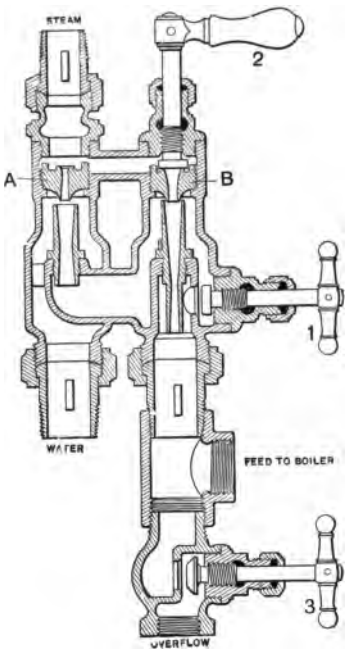

Fig. 80. Hancock Inspirator.

jectors of this type are called automatic, because they will re-establish the jet entering the boiler after it has been temporarily interrupted from any cause.

Types of Injectors.—Fig. 78 shows a section through the Lunkenheimer automatic injector, the corresponding parts being lettered the same as in Fig. 77. The check valve mentioned above is shown at *C*. The method of making the pipe connections for an injector of this type is indicated in Fig. 79.

These injectors are started by opening the valves *B*, *J* and *L* in succession. If the water is taken from a well, the valve *G* in the suction may remain open if the boiler is fed at frequent intervals.

Non-automatic injectors have the overflow closed by a valve instead of a check, which has to be operated by hand in starting and stopping. The Hancock inspirator shown in Fig. 80 is of this type, having a lifting jet *A* and a forcing jet *B*. To start the inspirator, open the overflow valves 1 and 3 and close the forcing steam valve 2. Then open the starting valve in the steam pipe. This puts the lifting jet in operation, the water being discharged through the overflow. Next close valve 1, and open 2 a quarter turn and close 3. This admits steam through the forcing jet *B*, and, the overflow valves being closed, the water is fed into the boiler. Table XLIII. gives the capacities of this type of injector for different sizes.

The range of the "Stationary" type of Hancock inspirator under different conditions is given in Table XLIV. These figures are for the regular patterns. Special inspirators are made

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which will operate on less than 15 pounds pressure. When used on a short lift they will take feed water up to 150° F. with steam pressures from 80 to 150 pounds.

There are many reliable makes of injectors upon the market, both of the automatic and non-automatic type. Those described above have been taken as simple forms well adapted to illustrate the principles upon which they operate. Fig. 81 shows the method of connecting an injector into the discharge pipe from a feed pump.

TABLE XLIII.*

CAPACITIES OF HANCOCK INSPIRATOR AND SIZES OF PIPE CONNECTIONS.

Size (number)	Capacities per hour		Pipe connections			
	With 60 lb. steam pressure and 4 ft. lift	Maximum horse power	Steam	Suction	Delivery	Overflow
7½	60 gals.	8	⅜	⅜	⅜	¼
8¾	90 "	12	⅜	½	½	⅜
10	120 "	16	⅜	½	½	⅜
12½	220 "	30	½	¾	¾	½
15	300 "	40	½	¾	¾	½
17½	420 "	56	¾	1	1	¾
20	540 "	72	¾	1	1	¾
22½	720 "	96	1	1¼	1¼	1
25	900 "	120	1	1¼	1¼	1
30	1,260 "	168	1¼	1½	1¼	1¼
35	1,740 "	230	1¼	1½	1½	1¼
40	2,230 "	298	1½	2	2	1½
45	2,820 "	376	1½	2	2	1½
50	3,480 "	464	2	2½	2½	2
55	3,650 "	600	2	2½	2½	2

TABLE XLIV.*

RANGE OF HANCOCK INSPIRATORS.

Lift	Minimum steam pressure	
	Feed-water cold	Feed-water at 100 degrees F.
5 feet	15 pounds	15 pounds
10 "	20 "	30 "
15 "	25 "	40 "
20 "	35 "	45 "
25 "	45 "	50 "

* From the catalogue of the Hancock Inspirator Co.

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Damper Regulators

Every boiler plant should be provided with automatic means for closing the dampers when the steam pressure reaches a certain point, and for opening them again when the pressure drops.

Hydraulic Regulators.—For power work and other cases where a close and reliable regulation is required, hydraulic regulators are used. The Locke, shown in Fig. 82, illustrates the general

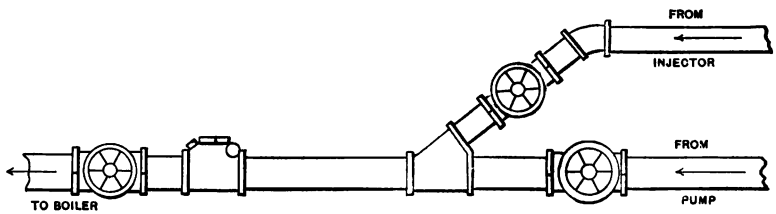

Fig. 81. Connecting Injector with Feed Pipe.

principles upon which this type of regulator operates. In this case the steam pressure is admitted beneath a metal diaphragm which is so weighted as to rise at the limiting pressure it is desired to carry on the boiler. This movement of the diaphragm operates a lever which in turn opens a small valve and admits water pressure beneath a piston. The rising of the piston closes the damper by means of a chain and pulley attachment as indicated. A special cut-off arrangement is provided by means of which the damper may be made to take any intermediate position necessary to give the right amount of draft to maintain an even pressure.

There are several regulators operating upon the same general principle although differing more or less in form and construction.

Blow-off Arrangement

Blow-off Tanks.—Where the blow-off from a boiler discharges into a sewer, some means must be employed for cooling the water or else the joints of the sewer pipe will be injured. This is accomplished by first passing the water through a special chamber or receiver, called a blow-off tank, one form of which is shown in Fig. 83. This consists of a cast iron receiver *A*, connected with the boiler through the blow-off pipe *B*. The tank ordinarily stands full of cold water.

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When hot water is admitted from the boiler, any steam which is formed will be carried off through the vapor pipe *C*, which should extend through the roof of the boiler house. As the hot water enters the top of the tank, the cold water will be forced out from the lower part through the discharge pipe *D*, which con-

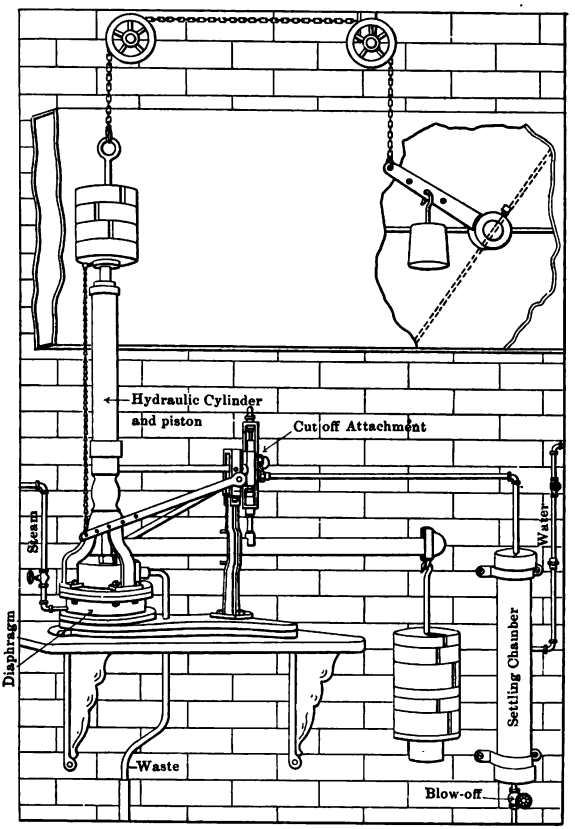

Fig. 82. Locke Hydraulic Regulator.

nects with the sewer. A small cross-connection *E* is provided for admitting air pressure to the discharge pipe to break the siphon effect and prevent the tank from being drained after the valve in the pipe *B* is closed. This form of tank is usually sunk in the ground so its top is flush with the boiler-room floor, or slightly above it.

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Fig. 84 shows a tank of boiler iron made to rest on cast-iron cradles or brick piers. The reference letters in this case indicate the same parts as in Fig. 83. There are in addition a pipe *F* for draining the tank, and a cross-connection *G* for use when the bottom of the boiler is lower than the top of the tank. A manhole should be placed in the shell on top, and a handhole in each end.

Tanks of this description are usually made of iron or steel one fourth inch thick with single riveted joints.

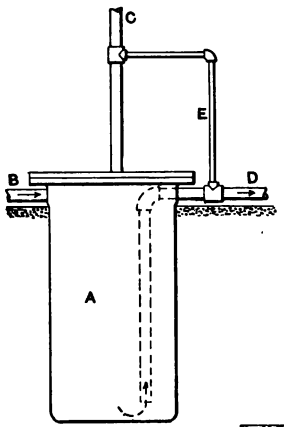

Fig. 83.

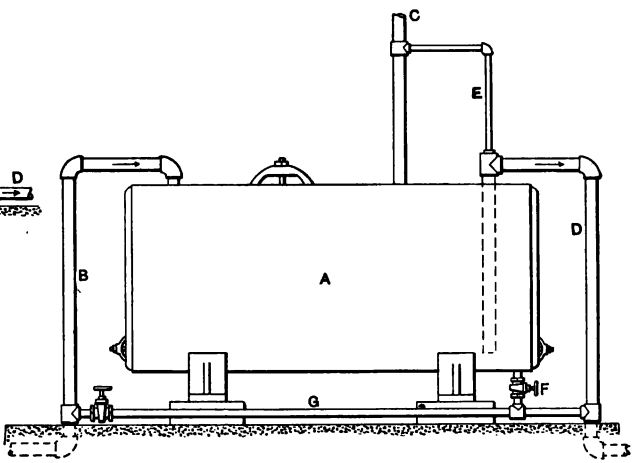

Fig. 84.

Blow Off Tanks.

The size of the blow-off tank depends upon the amount of water it is desired to remove from the boiler at one time. It is customary to blow off a small quantity at frequent intervals, and for this purpose it is usually found sufficient to make the capacity of the tank about one twenty fourth that of the total volume of the boiler neglecting the tubes. If there are two boilers, the tank should be made twice as large as for one, or two tanks may be used in series. This holds true up to the maximum number of boilers it is desired to blow off at one time. In a battery of several boilers they would probably be blown off in sections of two or three, and the tank may be proportioned accordingly.

Arrangement of Blow-off Pipe.—The blow-off pipe is tapped into the lowest point at the rear end of the boiler or into the mud drum in special forms. A gate valve and asbestos packed cock

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are connected into the pipe, outside of the setting, the valve being placed next to the boiler. Special blow-off valves, described in a later chapter, are often substituted for the cock in high pressure work.

Different methods are used for protecting the blow-off pipe where it is exposed to the fire in the combustion chamber. In

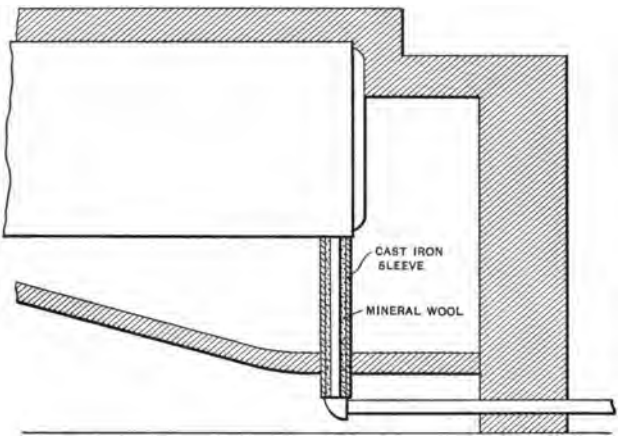

Fig. 85. Arrangement of Blow-Off Pipe.

most water-tube boilers it is located well away from the hottest part of the fire so that it is only necessary to consider here the blow-off pipe of a tubular boiler. If the blow-off and connections

TABLE XLV.

SIZES OF BLOW-OFF CONNECTIONS FOR TUBULAR BOILERS.

Diameter of boiler	Size of blow-off pipe	Size of pipe for gravity return
In.	In.	In.
36	1¼	1½
42	1½	2
48	1½	2
54	2	2½
60	2	2½
66	2½	3
72	2½	3

are exposed directly to the hot gases as they pass over the bridge wall they are liable to become overheated and give way. This is especially the case with power boilers where there is no circulation

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through the pipe. With heating boilers where the returns are brought back through the blow-off pipe a circulation is set up which greatly lessens this danger.

Fig. 85 shows a good arrangement in which the pipe is carried down through the paving and then outside the setting as indicated. The part exposed to the fire is encased in a split cast-iron sleeve held in place by special clamps and bolts. The space between the pipe and sleeve should be packed with some non-conducting material like asbestos or mineral wool.

Table XLV. gives suitable sizes of blow-off connections for boilers of different diameters. When the pipe acts also as a gravity return for the condensation from a heating system it is commonly made a size larger.

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CHAPTER XII.

STEAM ENGINES AND TURBINES

Work.—A force or pressure acting through space produces work.

The unit of work is the *foot pound,* and is the work required to raise a weight of 1 pound 1 foot in height. Thus, to raise 100 pounds 50 ft. requires $100 \times 50 = 5{,}000$ foot pounds. Neither the character of the force acting nor the direction of the movement has any effect upon the result so far as the amount of work done is concerned, provided the product of the force, times the distance moved through, is the same in each case.

Power is the rate of doing work, or the amount done in a specified time. The unit of power is the *horse power,* which is equivalent to 33,000 foot pounds per minute.

The Steam Engine.—In a steam engine steam is the agent by means of which heat is transformed into mechanical work. It is customary to speak of obtaining work from steam, but strictly speaking, it is the heat contained in the steam which produces work, and not the steam itself.

The essential features of a steam engine are a closed cylinder fitted with a piston and rod, and a valve, or series of valves, so arranged as to alternately admit and exhaust steam from each end of the cylinder. In operation, high-pressure steam is admitted to one end of the cylinder, and exerting a pressure upon the piston, pushes it forward, forcing out the low-pressure steam in front of it through the exhaust valve.

At the end of the stroke the admission and exhaust valves are reversed, and the operation is repeated on the return stroke.

The piston moves because the pressure on one side is greater than on the other, and in order to move, work must be done. The amount of work is easily computed, since it equals the total pressure acting on the piston, multiplied by the distance through which it moves. For example, a piston having an area of 3 square feet is acted upon by a steam pressure of 75 pounds per square inch, gauge pressure, through a stroke of 4 feet. What is

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the amount of work done per stroke, if the other side of the piston is open to the atmosphere? The total pressure acting upon the piston is $3 \times 144 \times 75 = 32{,}400$ pounds, and the work done equals this force multiplied by the distance through which it acts, or $32{,}400 \times 4 = 129{,}600$ foot pounds.

Work Diagram.—The action just described may be represented graphically by the diagram shown in Fig. 86, called a work diagram. This is constructed by drawing two lines *O X* and *O Y* at right angles to each other, and letting distances measured on *O X* represent space moved through by the piston, and distances measured on *O Y*, the pressures acting. The pressure or force

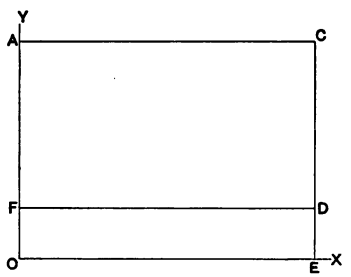

Fig. 86. Work Diagram.

available for doing work in any case is the difference between the two pressures acting upon the opposite sides of the piston and is called the *effective* pressure. In computing the work done by an engine exhausting into the atmosphere, we must deduct the atmospheric pressure from the *total* steam pressure to obtain the *effective* pressure.

Returning now to the diagram, let us apply it to the example previously given.

First lay off on an assumed scale a distance *O E* to represent the stroke of the piston, which is 4 feet. Next lay off to scale *O F*, equal to the atmospheric pressure per square foot acting on the piston, or $15 \times 144 \times 3 = 6{,}480$ pounds; and *O A*, equal to the total steam or boiler pressure, which is $(75 + 15) \times 144 \times 3 = 38{,}880$ pounds. From the definition given, the total work per stroke is equal to the pressure *O A*, multiplied by the distance *O E*, which is evidently the area of the rectangle *O A C E*. The effective pressure acting is the difference between the total and

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the atmospheric pressure, or that represented by the line *F A;* and the effective work done is that shown by the area *F A C D.* This rule holds true whatever the shape of the diagram.

If the pressure varies during different parts of the stroke so that the line *A C* is irregular or sloping, the enclosed area still represents the work done. In our consideration of Fig. 86, steam was admitted to the cylinder during the entire stroke, but in actual practice this is rarely done. It is customary, instead, to admit

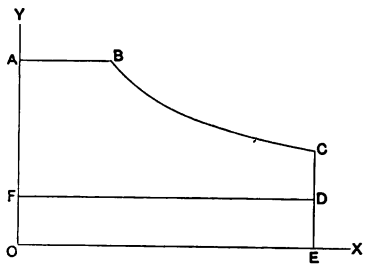

Fig. 87. Work Diagram.

steam during only a part of the stroke and then allow it to expand, as the piston moves forward, until it fills the entire volume of the cylinder. This is shown in Fig. 87.

Steam is admitted in this case until the piston reaches a point *B,* corresponding to one third of its stroke. Communication with the boiler is then cut off and the steam expands until it fills the cylinder, pushing the piston before it. Since there is the same weight of steam present at every point in the stroke, while the volume continues to increase, it follows that the pressure must decrease correspondingly; that it, the product of the pressure, multiplied by the volume at all points of the stroke beyond the point of cut-off, is the same in each case.

This is shown in Fig. 87. The line *A B* is horizontal because the pressure remains the same to the point of cut-off. The pressure then begins to fall, and is represented by the curved line *B C* for the remainder of the stroke.

In this case, as before, the work done is represented by the area *F A B C D.*

Mean Effective Pressure.—Whatever the form of the indicator diagram, we may always construct a rectangle having the same

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area and length, if we know the area and length of the diagram, for the height may be found by dividing the area by the length.

The height of this equivalent rectangle is called the *mean ordinate,* or *mean effective pressure* when applied to the indicator diagram of a steam engine.

Back Pressure.—In the example given, a back pressure was assumed equal to that of the atmosphere (15 pounds per square inch), but in practice this will vary according to circumstances. If the engine exhausts into a condenser, it will be much less, and if it exhausts into the open air a back pressure will be produced somewhat greater than atmospheric pressure, due to the friction of the steam in passing through the ports and exhaust pipe.

The *effective* pressure at any given point in the stroke is the difference between the total or boiler pressure and the back pressure, and the *mean effective* pressure is the average, or mean, of the effective pressures acting at all points throughout the stroke.

Indicator Diagram.—If steam were admitted to the cylinder at boiler pressure during the full stroke, the work done could be easily computed, since it would equal the pressure per square inch acting upon the piston, multiplied by the area of the piston in square inches, times the length of stroke in feet. We have seen, however, that the pressures upon both sides of the piston may vary under different conditions, so that in order to determine the power of an engine it is necessary to obtain a diagram like that shown in Fig. 87, giving the pressures upon both sides of the piston for all parts of the stroke.

The actual work diagram from an engine is obtained by the use of an instrument called an indicator, and the diagram is called an *indicator diagram* or card. In principle an indicator consists of a small cylinder containing a piston and rod. Steam pressure from the engine cylinder is admitted below the piston, which has a stiff spring above it. A pencil is attached to the end of the piston rod with its point resting against a paper stretched over a metal drum, which, in turn, is connected by means of a cord with the crosshead of the engine. In practice it would be impossible to represent pounds pressure by feet, or to show the actual stroke of the engine upon a diagram, so its size is reduced

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by attaching a reducing motion to the indicator drum, which makes the actual diagram about 3 or 3½ inches in length.

The vertical dimension of the diagram is reduced by placing a spring above the piston as already stated. These springs are usually designated as 10-pound, 20-pound, 40-pound, etc. This means that with a 10-pound spring the pencil point will be raised one tenth of an inch for each pound pressure per square inch in

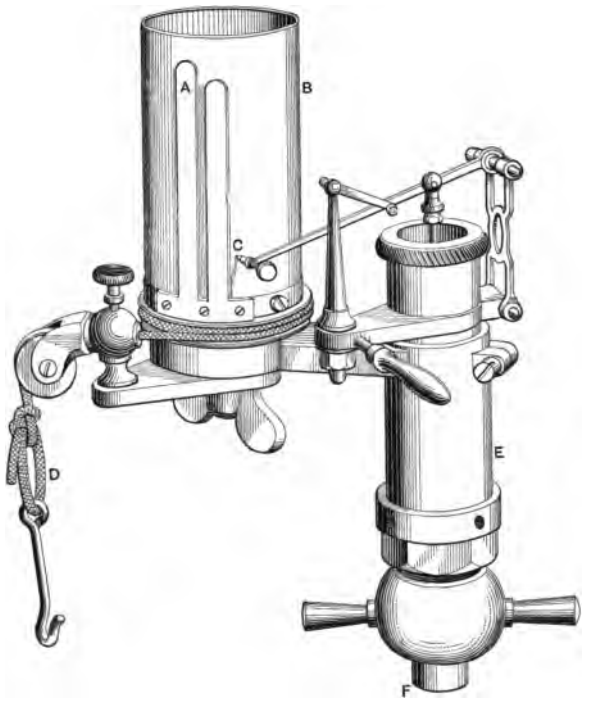

Fig. 88. Steam Engine Indicator.

the cylinder, or each 10 pounds pressure will raise the pencil 1 inch. With a 20-pound spring, 1 pound pressure will raise the pencil one twentieth of an inch, and so on for any other strength of spring. Fig. 88 shows a common form of indicator.

To obtain a diagram, or card, a strip of blank paper is stretched around the drum of the indicator and the actuating cord attached to the crosshead of the engine; this gives a forward and backward

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motion to the cord corresponding to that of the engine piston. The cock connecting the indicator cylinder with the engine cylinder is now opened and the pencil at the end of the small piston rod moves up and down distances corresponding to the pressures in the engine cylinder during different parts of the stroke. By pressing the pencil point against the card upon the drum, a *pressure line* will be traced corresponding to the line *A B C* in Fig. 87, which shows the pressures in the cylinder at all parts of the stroke.

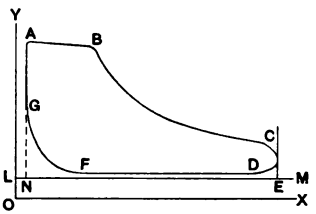

Fig. 89. Indicator Diagram.

When taking an actual card from an engine it will be found to differ somewhat from the theoretical diagram taking a form more nearly like that shown in Fig. 89.

The rounding of the corners is due to the relatively slow action of the valves and to the compression of a small amount of steam for cushioning purposes on the return stroke. Owing to the resistance to the flow of steam through the various passages, the back-pressure line will be found to be slightly above that of the atmospheric line in case of a non-condensing engine.

The same variation from the absolute vacuum line will be found on cards taken from a condensing engine, due partly to the same reason and partly to the imperfect action of the condenser.

Referring to Fig. 89, which represents a diagram from a non-condensing engine *L M* is the atmospheric line, drawn when the indicator is shut off from the engine cylinder, *G A* is the admission line, taken when steam is first admitted to the cylinder and showing the rapid rise in pressure at this time. The steam line *A B* is drawn during the period of admission, and usually falls a little, due to "wire-drawing" through the valve and port. The valve closes at *B*, and the expansion line *B C* indicates the falling pressure in the cylinder due to expansion as the piston moves forward.

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The exhaust valve opens at the point *C*, and the pressure falls immediately after, as shown by the line *C D*. The line *D F* represents the back pressure during the return stroke, which in case of a non-condensing engine, is slightly above the atmospheric line *L M*. At *F*, near the end of the return stroke, the exhaust valve closes, retaining a certain amount of steam in the cylinder for cushioning purposes, the effect of which is shown by the compression line *F G*.

The line of zero pressure, or line of absolute vacuum *O X*, is drawn below and parallel to the atmospheric line a distance to scale, represented by 15 pounds pressure per square inch.

Clearance.—Clearance is the waste space at the end of the cylinder between the head and the piston when the latter is at the end of its stroke, including the counter-bore and the ports up to the face of the closed valve. It is usually expressed as a percentage of the total piston displacement and commonly varies from 8 to 12 per cent in high-speed engines, and from 2 to 3 per cent in low-speed.

The clearance line *O Y* in Fig. 89 is drawn perpendicular to the line of absolute vacuum, and at a distance from the end of the diagram equal to the percentage of the clearance volume expressed as the same percentage of the stroke.

Computing the Power of an Engine

To accurately determine the horse power of an engine working under given conditions it is necessary to take several indicator diagrams or cards from both ends of the cylinder, then compute the work from an equal number of each and take the average of the lot.

The work per stroke is computed from an indicator card as follows: The area of the diagram, in square inches, is first measured with an instrument called a planimeter. This area is divided by the length of the diagram, in inches, to find the mean ordinate, which, multiplied by the number of the spring, will give the mean effective pressure (M. E. P.). The M. E. P. multiplied by the area of the piston in square inches, times the length of stroke in feet, gives the foot pounds per stroke. A horse power (H. P.) has already been defined as work done at the rate of 33,000 foot pounds per minute. Therefore, the work per stroke,

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in foot pounds, multiplied by the number of strokes per minute, divided by 33,000, equals the horse power.

Example.—An engine running at 300 revolutions per minute has a piston 12 in. in diameter and a 24-in. stroke. An indicator diagram taken with a 60-lb. spring has a length of 3 in. and an area of 2.4 sq. in. What is the horse power of the engine?

Mean ordinate $= 2.4 \div 3 = .8$.

M. E. P. $= .8 \times 60 = 48$.

The area of the piston is 113 sq. in., therefore

$$\frac{48 \times 2 \times 113 \times 600}{33{,}000} = 197 \text{ H. P.}$$

Rule for Horse Power.—The whole operation may be stated in the form of a rule as follows:

"Find the mean effective pressure by the use of a planimeter, multiply this by the area of the piston in square inches. Multiply this result by the length of stroke in feet, and the product by the number of strokes per minute. The final product divided by 33,000 will be the horse power of the engine."

A simple way to state this rule is in the form of an equation as follows:

$$\text{H. P.} = \frac{P \times L \times A \times N}{33{,}000}$$

in which

P = M. E. P. in pounds per square inch.

L = length of stroke, in feet.

A = area of piston, in square inches.

N = number of *strokes* per minute.

This is easily remembered because the initials of the quantities in the numerator of the second member of the equation spell the word *PLAN*.

Engine Constant.—Where a large number of cards are to be worked out, and where the speed of the engine is practically constant, it is often convenient to first compute what is known as the "engine constant."

This is found by multiplying the area of the piston by the length of stroke, and this by the number of strokes per minute, and dividing the result by 33,000. This quantity multiplied by the

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M. E. P. of any diagram will give the horse power as indicated by that particular card, or if it is multiplied by the average M. E. P. of all the cards it will give the average horse power for the whole test.

Computing the Approximate Power.—The approximate power of an engine may be determined without the use of an indicator diagram if the percentage of clearance, the point of cut-off, and the boiler pressure are known.

In computing the horse power by the method just given, the M. E. P. is the only factor obtained from the indicator card. By

TABLE XLVI.

VALUES OF M. E. P. FOR DIFFERENT CONDITIONS.

Cut-off	Per cent clearance					
	0	1.75	3	5	7	9
1/16	0.330	0.354	0.369	0.392	0.413	0.432
1/8	0.465	0.481	0.492	0.509	0.524	0.538
1/4	0.597	0.607	0.615	0.626	0.636	0.646
3/8	0.699	0.707	0.713	0.719	0.727	0.734
1/2	0.766	0.771	0.775	0.781	0.786	0.792
5/8	0.846	0.850	0.852	0.856	0.858	0.862
3/4	0.966	0.966	0.966	0.967	0.968	0.969
15/16	0.995	0.995	0.995	0.995	0.995	0.995

the use of Table XLVI. this can be determined approximately for different conditions, and then substituted in the equation for horse power, after which the problem may be solved as before.

The ratios given in this table are for absolute pressures, and their use can best be shown by a practical example.

Example.—A non-condensing engine with 5 per cent clearance cuts off at 1/4 stroke; the boiler pressure is 75 lb. gauge. What is the M. E. P.?

A gauge pressure of 75 lb. $= 75 + 15 = 90$ lb. absolute. The ratio for 1/4 cut-off and 5 per cent clearance is 0.626; therefore the mean pressure above a vacuum is $90 \times 0.626 = 56.3$ lb.

Under ordinary conditions the back pressure on the piston of an engine exhausting into the atmosphere will be about 2 lb. above atmospheric pressure, or 17 lb. absolute, so to find the M. E. P. in the above example, we must subtract the back pressure from the mean pressure, which gives $56.3 - 17 = 39.3$ lb.

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The average pressure in a condenser may be taken as about 3 lb. absolute, so that briefly we may write, M. E. P. = $[(p + 15) \times \text{ratio}] - 17$, for non condensing engines, and M. E. P. = $[(p + 15) \times \text{ratio}] - 3$, for condensing engines, in which p is the boiler gauge pressure.

Effect of Change in Cut-off and Back Pressure.—Table XLVI. is also useful for determining the effect upon an engine of changing the point of cut-off, or of increasing or decreasing the back pressure.

Example.—A non-condensing engine having a clearance of 3 per cent cuts off at 1/6 stroke when running under a boiler pressure of 85 lb. gauge. It is desired to increase the power of the engine 40 per cent by increasing the cut-off, all other conditions remaining the same. What will be the required point of cut-off? Absolute boiler pressure = 85 + 15 = 100 lb. The mean pressure ratio for 3 per cent clearance and 1/6 cut-off is 0.492, from which M. E. P. is found to be $(100 \times 0.492) - 17 = 32.2$ lb. As the power of an engine varies directly as the M. E. P., all other conditions remaining constant, the required M. E. P. will be $32.2 \times 1.40 = 45$ lb.

Now reversing the operation, we find the mean pressure ratio to be $(45 + 17) \div 100 = 0.620$. Looking in Table XLVI. in the column for 3 per cent clearance, we find that a ratio of 0.615, which is very close to that required, corresponds to a cut-off of ¼, which is the one to be used.

Example.—Suppose in the above case it was desired to use the exhaust steam for drying purposes under a pressure of 10 lb. gauge. What would be the resulting loss in power?

The M. E. P. for the engine cutting off at ¼ stroke and exhausting into the air was found to be 45 lb. Raising the back pressure to 10 lb. gauge or 25 lb. absolute, means a reduction in M. E. P. of $25 - 17 = 8$ lb.; thus giving an M. E. P. of $45 - 8 = 37$ lb. under the required conditions. Dividing 37 by 45 gives 0.82, which means that the resulting M. E. P., and consequently the power of the engine, would be only 82 per cent of what it was originally; that is, the power has been reduced $100 - 82 = 18$ per cent.

If it were necessary to keep the power the same by increasing the point of cut-off, we should proceed as in the previous exam-

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ple, except for an increase of 18 per cent in power instead of 40 per cent as before.

If it were desired to keep the cut-off at ¼ and bring the engine up to its original rating by raising the boiler pressure, we should proceed as follows:

Required M. E. P. = 45 lb.

Back pressure = 25 lb.

Total mean pressure = 45 + 25 = 70 lb.

Mean pressure ratio from Table XLVI. is 0.615.

Required absollute boiler pressure is 70 ÷ 0.615 = 113 lb., or an increase of 113 — 100 = 13 lb. is required in the boiler pressure.

Mechanical Efficiency.—The preceding methods for computing the power of an engine give what is known as the *indicated* horse power (I. H. P.), and include the work required to overcome the friction of the engine itself.

The *delivered* or *brake* horse power (D. H. P.) is less than the indicated horse power, usually varying at full load from 80 to 90 per cent, depending upon the size and construction of the engine. The ratio of the D. H. P. to the I. H. P. is called the *mechanical efficiency,* and, in general, increases with the size of engine.

As the load upon any given engine is reduced, the efficiency becomes less; this is true because the friction, remaining practically constant, becomes greater in proportion to the total load as the useful or delivered work becomes less.

Thermal Efficiency.—This is the ratio of the heat transformed into work, to the total heat supplied to the engine, and depends upon the difference in temperature of the steam at initial and exhaust pressures. It is equal to

$$\frac{T_1 - T_2}{T_1}$$

in which T_1 and T_2 are the absolute temperatures of the steam at initial and exhaust pressures respectively.

Example.—What is the thermal efficiency of an engine taking steam at an initial pressure of 100 lb. gauge, and exhausting into the atmosphere under a pressure of 2 lb.

The absolute temperature of steam at 100 lb. gauge pressure is

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$337 + 461 = 798°$, and that of steam at 2 lb. pressure is $219 + 461 = 680°$. Therefore the efficiency is

$$\frac{798-680}{798} = 15\%, \text{ nearly.}$$

If the engine should be connected with a condenser, the range of temperature worked through would be greater, and the efficiency would be correspondingly increased. In actual practice this efficiency cannot be obtained because the difference between the amount of heat received and that rejected in the exhaust is not all converted into work; part of it being lost by radiation, conduction, leakage, cylinder condensation, etc. Its principal use is in showing the advantage to be gained by the higher ratios of expansion in connection with engines of proper construction.

Water Rate.—The water rate of an engine is the weight of steam required per I. H. P. per hour. This depends upon the cylinder losses already mentioned, condensation being by far the most important. During the period of exhaust the entire internal surface of the cylinder is exposed to steam at exhaust pressure, and a layer of iron of greater or less thickness is reduced to a temperature approximating that of the exhaust steam. When steam at boiler pressure is admitted at the beginning of the succeeding stroke, a certain amount of heat is given up in raising the temperature of the cylinder walls to that of the inflowing steam.

This results in the condensation of a given quantity of steam at each stroke which is a dead loss so far as useful work is concerned, but which must be included in the total amount supplied to the engine. Leakage around the valves, radiation, etc., add to what are commonly known as "cylinder losses." These vary with different types of engines, and with other conditions to be mentioned later. Cylinder condensation increases with the ratio of expansion, for the greater the difference between the initial and final temperatures of the steam, the greater will be the cooling effect on the cylinder between the exhaust and admission periods.

The amount of heat in the steam which is actually converted into useful work is very small compared with that rejected in the exhaust and lost by radiation and condensation. For example, 1 horse power represents 33,000 foot pounds of work per minute;

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and the heat equivalent of this is $33{,}000 \div 778 = 42 +$ B. T. U. per minute or 2,520 B. T. U. per hour. The latent heat of steam at 100 pounds gauge pressure is 876 B. T. U., or in other words, the available heat in $2{,}520 \div 876 = 2.8$ pounds of steam would be sufficient to generate 1 horse power if it could all be turned into useful work. Actually over 30 pounds of steam are required per horse power per hour in the average simple non-condensing engine, nine tenths of which is required to offset the heat carried off in the exhaust and dissipated in the various cylinder losses. In practice, cylinder condensation is reduced by the use of superheated steam and by compounding. Table IV., page 29, gives water rates for different types of engines.

Superheated Steam.—It has been stated that when steam is first admitted to the cylinder, at the beginning of the stroke, condensation takes place rapidly owing to the marked difference in temperature between the steam and the cylinder walls. Furthermore, experiment has shown that condensation is greatly increased if the metal surfaces are moist. As superheated steam has a temperature considerably higher than the point of saturation at initial or admission pressure, a certain amount of cooling may take place before condensation begins.

This has the effect of raising the temperature of the cylinder walls without moistening them, thus retarding the transfer of heat and practically eliminating condensation during the admission period. While this is the principal advantage arising from the use of superheated steam, it is also lighter for a given volume, and therefore a less weight is required to fill the cylinder up to the point of cut-off, thus reducing the water rate to a certain extent. The most desirable degree of superheat, for practical results, is that which will prevent condensation on the cylinder walls up to the point of cut-off.

The disadvantages of superheat are the cutting effect upon the valves and steam passages and the difficulty of proper cylinder lubrication at high temperatures.

These, however, may be overcome to a considerable extent by suitable valve design and construction and by employing cylinder oil of the highest grade.

As the pressure, and consequently the temperature, of the steam

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increases, the degree of superheat, which it is practicable to employ, is correspondingly decreased.

In the case of reciprocating engines the limit of economy is found to range from 100 to 150 degrees above the normal temperature of the steam, while with turbines which require no lubrication within the steam passages, the limit may be raised to about 200 degrees.

Multiple Expansion Engines

The most effective results in the reduction of condensation are secured through "compounding."

We have already learned that cylinder condensation is due, in the greater part, to the wide fluctuation of temperature of the interior surfaces, caused by the difference in temperature of the steam at initial and exhaust pressures. This limits the initial pressure under which it is economical to operate a simple engine, because after a certain point is reached the increased cylinder losses will more than offset the gain in power.

Reasons for Compounding.—If a given weight of steam is expanded through the same range of pressure in two cylinders, one exhausting into the other, instead of in a single cylinder, the resulting condensation will be considerably reduced.

This may best be shown by a practical illustration: A simple, or single-cylinder engine is supplied with steam at an initial pressure of 100 pounds gauge, and exhausts at a pressure 2 pounds above atmosphere. The range of temperature to which the interior of the cylinder is exposed is $337 - 219 = 118°$.

Now, if two cylinders are used instead of one, and so arranged that the steam is expanded from 100 pounds pressure down to 34 pounds in the first cylinder, then exhausted into the second, and expanded down to 2 pounds, the temperature range in the first cylinder will be $337 - 279 = 58°$, and in the second $279 - 219 = 60°$.

The decrease in temperature range in each cylinder causes a considerable reduction in the total condensation for the entire expansion.

This holds true to a certain extent as the number of cylinders is increased to three and four.

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Another advantage to be gained by multiple expansion is due to the fact that a comparatively small amount of fuel is required to raise the pressure from 80 or 90 pounds, the common practice for simple engines, to 120 or 140 pounds, which makes it possible to increase the range of expansion and thus increase the power at a relatively small cost.

It is not possible to take advantage of this with simple engines on account of excessive condensation, but by increasing the number of cylinders this obstacle is overcome and a gain is made in two ways.

Initial Pressures.—For compound engines the initial pressure may be raised to about 140 pounds, thus realizing a gain of 20 to 30 per cent due to the higher pressure and greater expansion.

For triple expansion engines the economical pressure is of course higher than for compound, and may be increased to advantage up to 180 pounds or more. The gain in this case over the compound engine will be from 5 to 10 per cent, and even more in some cases.

Ratio of Expansion.—The best ratio of expansion depends upon the type of engine and the initial steam pressure.

For high-speed to moderately high-speed engines, such as are commonly used for driving electric generators, the following may be taken as about the average for the pressures carried:

Simple engines—3 to 4 expansions (cut-off 1/3 to 1/4).

Compound engines—8 to 10 expansions (cut-off 1/8 to 1/10).

Triple expansion engines—12 to 15 expansions (cut-off 1/12 to 1/15).

Types of Engines

Steam engines may be classified in different ways according to their mechanical construction, although there is no definite line of distinction.

They are designed to operate at both high and low speeds, are made vertical and horizontal, and are constructed with either single or multiple cylinders, according to the requirements. They may, however, in a general way, be included under the heads of slow-speed, moderate-speed, and high-speed, and will be taken up in this order.

Slow-speed Engines.—These are commonly characterized by

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speeds of 100 revolutions per minute or under, a stroke of 36 to 72 inches, independent steam and exhaust valves, the former being of the Corliss or other drop cut-off type, and are regulated by means of a fly-ball governor.

They are well adapted to factory work and for driving belted dynamos. The larger sizes are used to some extent with direct-connected machines, the latter being especially designed to operate at slow speeds.

Condensation is greater in an engine when runing at slow speed than when running at high speed, because the steam remains in contact with the cylinder walls for a greater length of time at each stroke. In the case of slow-speed engines, however, this is usually more than offset by the smaller clearance and more efficient valve gears which may be employed under these conditions, so that, as a class, this type of engine is more efficient than the high-speed.

Moderate-Speed Engines.—Following, and to some extent overlapping, the preceding type, are the medium-speed engines. These commonly run at a speed of 100 to 175 revolutions per minute, have a stroke of 24 to 42 inches, four valves positively driven, either by one or two eccentrics, side-crank construction, and automatic shaft-governor. They are adapted to locations where lack of space prohibits the use of a slow-speed engine, and where for any reason an excessively high speed would be undesirable. They are used for both power and lighting service, and may be directly connected with generators of suitable design.

High-Speed Engines.—These in most cases have a single valve, usually some modification of the slide valve, connected directly with the eccentric and controlled by an automatic shaft-governor. High-speed engines have come into general use of late years for driving electric generators, on account of the desirability of connecting the generator directly with the shaft of the engine.

This change has made it necessary to modify the general design somewhat, in order to overcome the effects of higher speeds.

The reciprocating parts must be made lighter to reduce vibration, and must be more carefully proportioned to maintain a proper balance. The bearings must be made much larger, and special care taken to keep them in adjustment.

In slow-speed engines the oil cups are easily watched, and any

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part of the engine can be oiled while running. This is impossible with those of high speed, and special means must be provided for supplying plenty of oil to the bearings and cylinder. Forced lubrication is often employed, with the running parts enclosed to prevent the throwing of oil.

This type of engine is made very compact, and with a short stroke as compared with those of slower speed.

Simple Engines.—This name is applied to those in which the total expansion of the steam takes place in a single cylinder. They may have one or more cylinders, but in the latter case both take steam at boiler pressure and exhaust into the atmosphere or condenser as the case may be. Simple engines cost less per horse power than compound, and are not so complicated as the latter; but on the other hand, they are more wasteful in the use of steam.

Compound Engines.—Compound engines commonly have two cylinders, so arranged that steam first enters the high-pressure cylinder, expands, and then exhausts into the low-pressure cylinder, where the expansion is completed.

Some engineers proportion the cylinders so that approximately the same amount of work is done in each, while others claim it is better to make the temperature range of the steam more nearly equal. There are various rules for determining the proper ratio between the cylinders, a simple one being to make it equal to the square root of the total ratio of expansion. For example, if the ratio of expansion is 9, the ratio of the cylinder volumes will be $\sqrt{9} = 3$; that is, the volume of the low-pressure cylinder should be three times that of the high-pressure cylinder.

The most common forms of the compound engine are the *tandem* and *cross-compound.*

In the first arrangement both pistons are placed upon the same rod (see Fig. 94), with the axes of the cylinders in line. Only one set of reciprocating parts is required, and except for the two sets of cylinders, pistons and valves, the appearance is the same as that of a simple engine.

Cross-compound engines are made up of two complete machines, except the main shaft and fly-wheel, which are common to both. One advantage which this form has over the tandem engine is that the cranks may be set at 90 degrees from each

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other, so that when one is on a dead center the other is at a position of nearly its greatest effort. This makes a dead center impossible and gives a more uniform turning moment.

When the cranks are set in this manner the low-pressure piston is not ready to receive steam when the high-pressure exhausts. This condition is overcome by placing a *receiver* between the two cylinders, into which the high-pressure cylinder exhausts and from which the low-pressure takes its supply. This is sometimes done away with by placing the cranks either in the same position or opposite, so that the strokes begin and end together. In this case the steam exhausts from the high-pressure cylinder

Fig. 90. Simple High-Speed Engine.

directly into the low-pressure; but with this arrangement the advantage of a uniform turning moment is lost.

Cross-compound engines are more expensive to make, and require a larger floor space than tandem engines, while the labor of cleaning and oiling is nearly doubled. On the other hand, the individual parts may be made lighter and are thus more easily handled.

Triple Expansion Engines.—In engines of this form the steam is expanded in three stages instead of two. Three cylinders are usually employed, the high, intermediate, and low, with the cranks set 120 degrees apart.

Sometimes four cylinders are used instead of three, the volume of the low-pressure cylinder being divided into two, which gives

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a better balance. Triple expansion engines are more frequently employed in marine work and large pumping stations, and are not well adapted to the conditions found in isolated plants, being more expensive and complicated than would be warranted by the gain in economy.

Descriptions of Engines of Different Types

The Harrisburg Standard engine, shown in Fig. 90 is a good illustration of a simple high-speed engine. This is shown with a direct-connected generator.

TABLE XLVII.

DIMENSIONS AND FLOOR SPACE OF HARRISBURG STANDARD BELTED TYPE ENGINES.

Simple non-condensing, 80 pounds pressure, ¼ cut-off

Diameter of cylinder	Stroke	Revolutions per minute	Indicated horse power	Diameter of steam pipe	Diameter of exhaust pipe	Floor space belted type	
In.	In.			In.	In.	Ft. In.	Ft. In.
5	8	425	15	2	2	3 10 by	6 10
7	8	425	26	2½	2½	3 10 "	6 10
8	9	400	35	3	3	3 10 "	7 0
9	10	350	45	3	3	4 8 "	7 10
11	10	350	65	3½	3½	4 8 "	7 10
12	11	325	80	4	4	4 8 "	8 0

Table XLVII. gives the power and the approximate floor space required by some of the different sizes of the belted pattern.

The horse powers given in the table are for 80 pounds initial pressure and ¼ cut-off, which may be taken as average conditions for engines of this type.

For determining the width of a direct-connected engine and generator, we may, in general, add the width of the "engine type" of generator to that of the belted engine. This is a safe rule to follow, but in many cases will give a dimension somewhat greater than the actual width. Table XLVII., and similar ones which follow, are for approximate use only, and should be supplemented by more exact data obtained from the makers. Neither do they give a complete list of the different sizes of the type of engine

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shown. Like the tables of dimensions given for special makes of boilers and other apparatus, they are furnished as guides to show the approximate space required for different designs, and are for the use of architects and engineers in proportioning the size of an engine room where the particular make of machine is not specified, but where the general *type* to be employed is known.

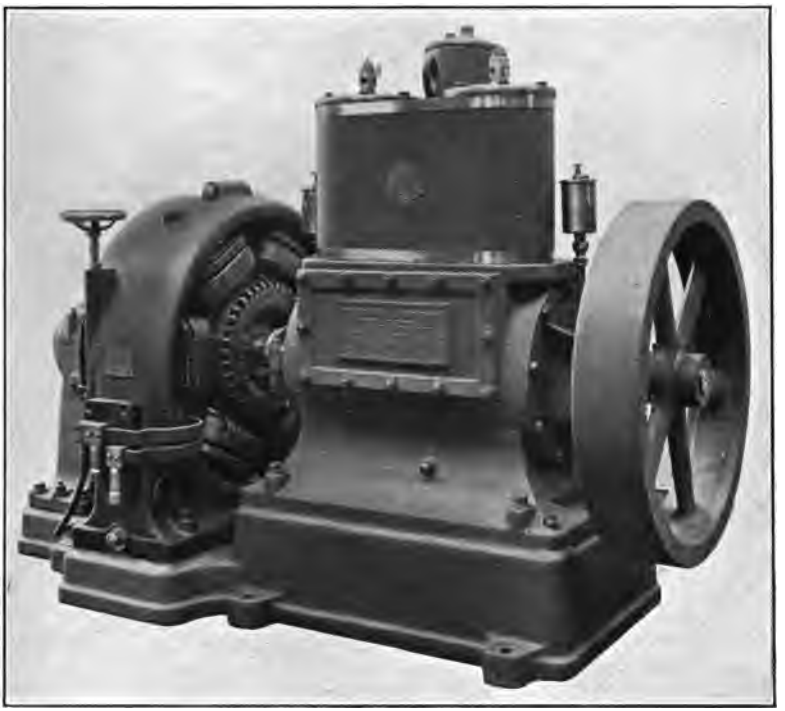

Fig. 91. Vertical High-Speed Engine.

Fig. 91 shows the Westinghouse Standard engine, with direct-connected generator. This is a vertical, single-acting, simple engine with two cylinders.

In single-acting engines, the steam is admitted only to the head end of the cylinder, the other being open to the atmosphere. The advantages claimed for this type of engine are that the pressure upon the moving parts is always in the same direction, and the bearings are less likely to wear loose, also the valve

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Fig. 92. Ball Medium Speed Engine.

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arrangement is simplified. Table XLVIII. gives a partial list of these engines with ratings at 80 pounds pressure. They may be connected with the generator in two ways. That shown in the cut is the more compact, one of the pulleys being removed and

TABLE XLVIII.

DIMENSIONS AND FLOOR SPACE OF WESTINGHOUSE ENGINES, BELTED TYPE.

Simple non-condensing, 80 pounds pressure

Diameter of cylinder	Stroke	Revolutions per minute	Indicated horse power	Diameter of steam pipe	Diameter of exhaust pipe	Floor space belted type	
In.	In.			In.	In.	Ft. In.	Ft. In.
6½	6	400	15	2	2	2 6 by	5 5
8½	8	400	35	3	3	3 0 "	6 8
11	10	330	60	4	4	4 0 "	8 0
13½	12	300	100	6	6	5 0 "	9 9
15½	14	300	150	6	6	6 6 "	11 0
20	16	250	250	8	10	6 6 "	11 6

TABLE XLIX.

DIMENSIONS AND FLOOR SPACE OF BALL MEDIUM-SPEED ENGINES, BELTED TYPE.

Simple non-condensing, 110 pounds pressure

Diameter of cylinder	Stroke	Revolutions per minute	Indicated horse power	Diameter of steam pipe	Diameter of exhaust pipe	Floor space belted type	
In.	In.			In.	In.	Ft. In.	Ft. In.
10	16	250	87	4	6	10–6 x	6–6
15	18	220	195	6	8	12–3 x	8–0
19	20	200	315	7	10	15–0 x	9–3
22	24	170	430	8	10	18–9 x	10–6
25	27	150	551	10	12	21–0 x	11–9
26	30	150	663	10	12	22–6 x	11–9
27	33	150	787	10	12	24–0 x	11–9

special bearings and supports provided to receive the generator upon the same bed-plate. In the second arrangement the generator is attached to the end of the shaft by means of a flexible coupling, and no change is made in the engine. The engine and

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Fig. 83. Slow-speed Simple Corliss Engine.

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generator may stand upon the same or separate bed-plates as desired.

The Ball engine shown in Fig. 92 is of the medium-speed type with Corliss valves arranged for positive action, and is combined with a direct-connected generator. Table XLIX. gives ratings and dimensions for several sizes of the belted machine.

Fig. 93 illustrates an engine of the slow-speed Corliss type, made by the Allis-Chalmers Company. These engines are used for driving dynamos as well as for factory and general power purposes; they may be either belted or direct-connected as conditions require. In the latter case, however, the speed is seldom reduced

TABLE L.

DIMENSIONS AND FLOOR SPACE OF CORLISS ENGINES, BELTED TYPE.

Simple non-condensing, 90 pounds pressure, ¼ cut-off

Diameter of cylinder	Stroke	Revolutions per minute	Indicated horse power	Diameter of steam pipe	Diameter of exhaust pipe	Floor space belted type, belted away from cylinder
In.	In.			In.	In.	Ft. In. Ft. In.
12	30	90	74	4	5	8 8 by 18 6
18	36	80	177	6	7	10 0 " 22 10
20	48	72	263	7	8	10 10 " 29 8
24	48	70	368	8	9	12 8 " 31 2
26	60	65	502	8	10	12 8 " 36 8
30	60	62	637	10	12	14 11 " 39 4

below 80 or 90 revolutions per minute. Table L. gives the rating of a few of the small and medium sizes working under 90 pounds steam pressure. The standard sizes run up to over 2,700 H. P. at 140 pounds pressure.

A typical form of the tandem compound engine is shown in Fig. 94, which illustrates the McEwen engine built by the Ridgway Dynamo and Engine Company.

Table LI. gives the rating of a number of different sizes working under a steam pressure of 100 pounds. The dimensions given are for the belted type, and are for cylinder ratios of 1 to 4.

For convenience in determining the approximate over-all dimensions of a direct-connected generating set, Table LII. may be used in connection with Tables XLVII. to LI., inclusive.

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Fig. 94. Tandem Compound Engine.

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Steam Turbines

Turbines, owing to their high speed and uniform efficiency under a considerable variation in load, are particularly adapted to the driving of electric generators, to which they may be directly connected. They require small foundations and take up little space, and owing to the small number of moving parts, the friction is slight.

Another advantage is the ability to use highly superheated steam, as no lubricating oil is required, and as the same parts of the interior are always exposed to steam of practically the same temperature, the loss from condensation is small.

TABLE LI.

DIMENSIONS AND FLOOR SPACE OF TANDEM-COMPOUND ENGINES, BELTED TYPE.

Compound non-condensing, 100 pounds pressure

Diameter of high pressure cylinder	Diameter of low pressure cylinder	Stroke	Revolutions per minute	Indicated horse power	Diameter of steam pipe	Diameter of exhaust pipe	Floor space belted type
In.	In.	In.			In.	In.	Ft. In. Ft. In.
8	16	12	275	75	3	7	5 0 by 12 8
9	18	14	250	105	4	8	5 6 " 15 1
10	20	16	225	135	4½	9	6 6 " 16 7
12	23	18	200	200	4½	10	7 0 " 18 11
14	28	20	180	250	6	12	7 6 " 19 6
16	32	22	165	350	7	14	8 6 " 20 6

The steam consumption of turbines is about the same as that of the best reciprocating engines, having been gradually reduced by various improvements in their construction.

Principle of the Turbine.—The principle upon which a turbine acts is different from that of a reciprocating engine. It does not derive its power from the static force of steam expanding behind a piston, but instead, the expansion produces a high velocity in a jet of steam, which is made to strike the vanes of a revolving wheel and thus give up its kinetic energy in turning the shaft and so producing work.

In both cases the work is due to the heat energy in the steam, the action being intermittent in the engine and continuous in the turbine.

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All turbines are included in two general classes according to the way in which the energy of the steam is utilized. The two types are commonly known as *impulse* and *reaction* machines, although both make use of the two principles of operation in varying proportions. The real distinction, however, is the manner in which

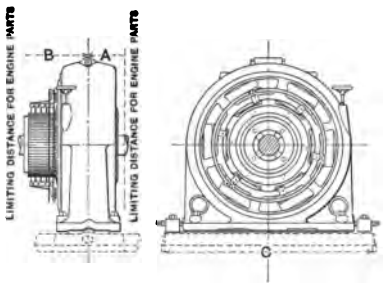

Fig. 95. Westinghouse Direct-Connected Generator.

TABLE LII.

FLOOR SPACE FOR WESTINGHOUSE ENGINE TYPE GENERATORS, DIRECT CONNECTED.

Kw.	Revolutions per minute	A	B	C
		In.	In.	In.
10	375 to 400	7	12	34
25	320 " 350	9½	18½	53½
50	275 " 300	10½	21⅜	63½
75	265 " 290	12½	26⅜	70
100	250 " 275	12½	27½	81½
150	200 " 225	15½	38	120
200	170 " 185	16	40	120
250	100 " 110	17	48	144

the steam is expanded, this being completed in the nozzle in the impulse turbine, while a portion takes place in the passages of the wheel in the reaction type.

Impulse Turbines.—Most of the machines made in this country make use of the impulse principle, so far as possible, in the development of power.

The heat energy of the steam in this case being first changed into kinetic energy by expansion in specially formed nozzles, after which the rapidly moving particles are discharged directly against the vanes or blades of the turbine wheel.

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The principle upon which the impulse turbine operates is illustrated in Fig. 96, in which *A* represents the steam chest, *B* the diverging nozzle, *C* the regulating valve, and *D* a section of the turbine wheel. In action the entering jet strikes the vanes, as

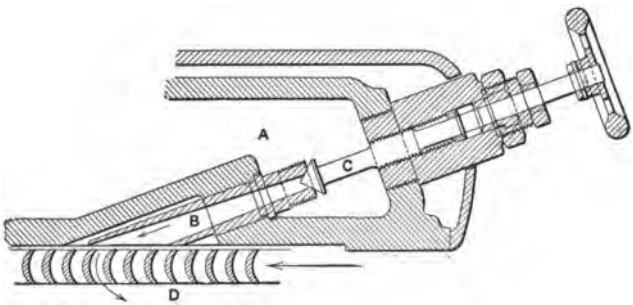

Fig. 96. Principle of Simple Impulse Turbine.

indicated by the arrows, and forces them forward, after which the direction of the steam is changed, and passes out at such

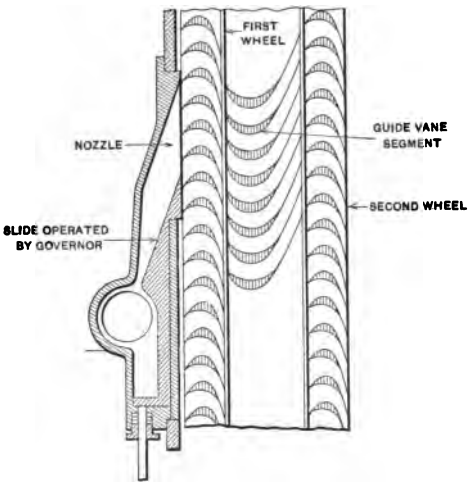

Fig. 97. Principle of Compound Turbine.

an angle with the axis of the wheel as to add a certain amount of reaction to the impulse effect already produced.

Compound Turbines.—The diagram shown in Fig. 96, repre-

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sents a *simple* turbine; that is, one having a single wheel, with a single set of blades.

The difficulty with this form is the excessively high speed necessary to utilize the energy of the steam. Turbines having two or more wheels attached to the same shaft, and separated by stationary guide vanes, are called *compound* turbines. Such a machine is shown in diagram in Fig. 97.

This arrangement results in a certain reduction of speed, as a part of the energy of the steam is imparted to each wheel. Theoretically, two wheels will reduce the speed to one half, and three wheels to one third that of a simple turbine, and so on, although this is not borne out exactly in practice.

In operation, the steam is first expanded in the nozzle, then passes through the first wheel, then enters the stationary guide vanes, which reverse the direction of flow, and finally enters the second wheel in the same manner as it did the first.

The arrangement of the fixed and movable blades is more clearly shown in Fig. 98, which represents a partial section through a turbine of this type. Referring to the cut, N is the nozzle, BB the wheel blades rotating together on the same shaft, and G the intermediate stationary blades for changing the direction of the steam.

Multiple Stage Turbines.—Impulse turbines are sometimes divided into stages as illustrated in Fig. 99.

In this arrangement the machine is made up of a series of turbines, each group of wheels and guides being placed in a separate compartment, but so connected that the exhaust from the first enters the nozzles of the second, and so on, according to the number of stages employed.

Reaction Turbines.—In the reaction turbine the steam is only partially expanded in the nozzle, being completed in the wheel itself.

A diagram of a compound reaction turbine having two wheels is shown in Fig. 100. This varies principally from the impulse type in the form of the vanes and in the substitution of guide vanes for nozzles.

The path of the steam, and the direction of rotation, are indicated by the arrows. As the steam enters, it first strikes the vanes in such a way as to impart a certain amount of pressure

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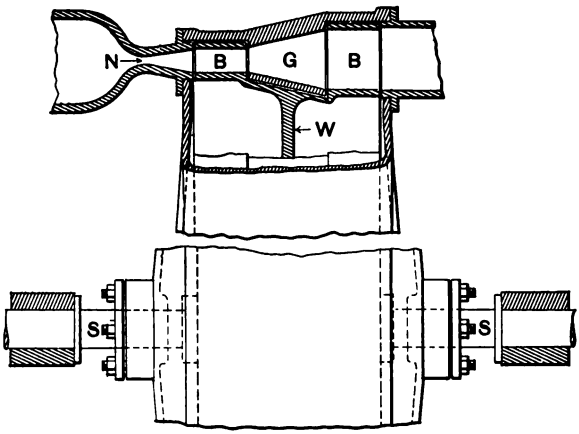

Fig. 98. Longitudinal Section of Blades, Single Stage Turbine.

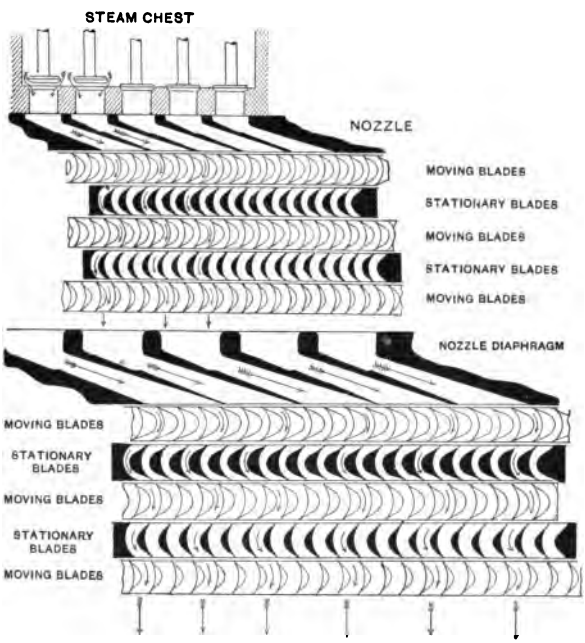

Fig. 99. Cross Section of Blades, Multiple Stage Turbine.

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by impulse; the direction is then changed by the angle of the blades and it leaves the wheel in such a manner as to react strongly, thus producing the larger part of the power developed in this way.

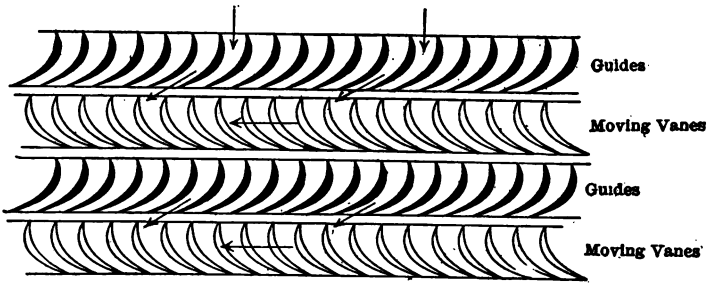

Fig. 100. Diagram of Compound Reaction Turbine.

Types of Turbines.—There are many different forms of turbines upon the market, of various sizes, and designed for different purposes. Space, however, will only be taken to show two or three typical designs.

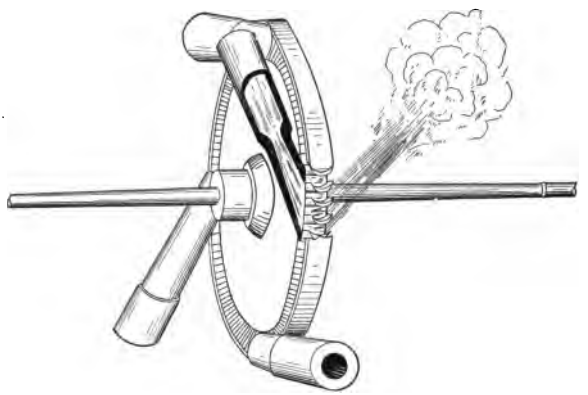

Fig. 101. De Laval Wheel and Nozzles.

De Laval Turbine.—This is a good illustration of a simple impulse machine, and is shown in diagram in Fig. 101.

In action, steam strikes the blades and is deflected; the impact of the jet and the reaction due to its deflection cause the wheel to rotate rapidly. The nozzles are small at the throat and

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diverge outward. By making them of the right length and with the proper amount of divergence the steam can be expanded from the pressure of admission to that of the condenser. Complete expansion is obtained in the diverging nozzle, and the steam leaves it at the exhaust pressure. The steam then works only by virtue of its high velocity.

This turbine has a long flexible shaft which can deflect enough to make up for any eccentricity of the center of gravity of the revolving parts.

Fig. 102. De Laval Generating Set.

The high speed of the wheel is reduced by means of especially constructed spiral gears running in oil inside of an enclosed casing. Speed regulation is obtained by means of a throttle valve in the admission pipe controlled by a fly-ball governor.

Fig. 102 shows the exterior of a De Laval steam turbine with direct-connected dynamos. The chamber surrounding the wheel is shown at the right; in the center is the casing containing the speed reducing gears, and at the left the dynamos.

Curtis Turbine.—This is of the compound impulse type and is made both vertical and horizontal in form, according to the size.

A horizontal two-stage machine of this make is shown in section in Fig. 103, and serves to illustrate the general method of construction. It will be noted on inspection that each of the two

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wheels has a double row of blades, with stationary guide vanes between them, and that the two stages or compartments are divided by a diaphragm.

In operation, steam first passes through nozzles leading from the steam chest and strikes the first set of blades upon the first

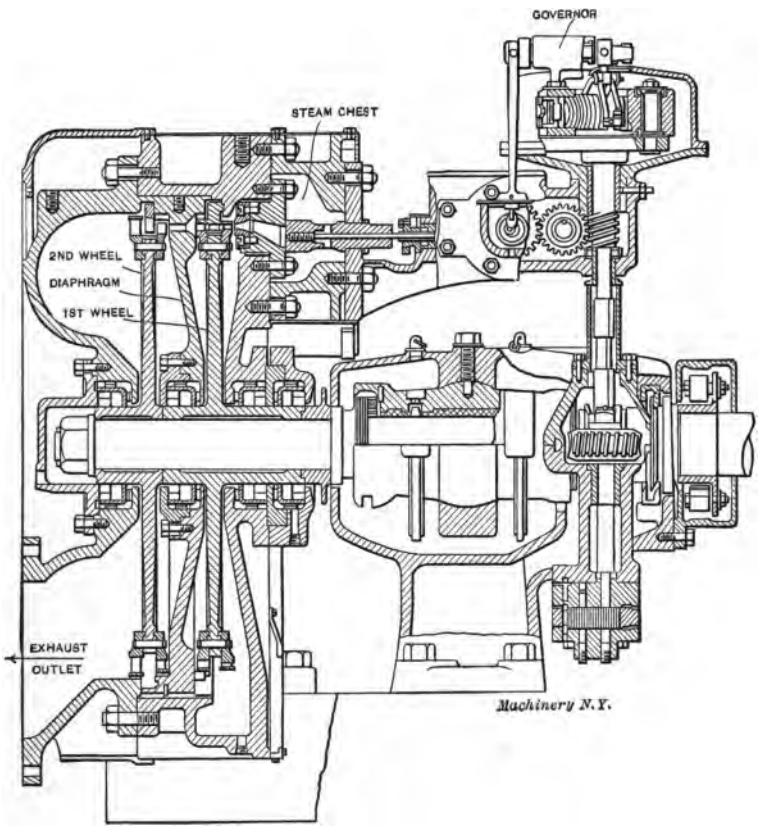

Fig. 103. Curtis Two-Stage Turbine.

wheel, then through the stationary vanes, which give it the proper direction for impinging upon the second set of blades upon the same wheel. The exhaust from the first wheel is admitted to the second compartment, where the operation is repeated in exactly the same manner, after which the steam passes into the exhaust outlet as indicated.

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Regulation is secured by means of automatically operated poppet valves which control the steam admitted through the nozzles. When less steam is required than will flow through a single nozzle, throttling is resorted to in one of them. Table LIII. gives over all dimensions of some of the standard sizes of the horizontal generating sets.

TABLE LIII.

OVER-ALL DIMENSIONS OF CURTIS TURBINES.

Rating	Voltage	Speed, revolutions per minute	Length	Weight, pounds
15 Kw.	85	4000	5 ft. 6 in.	1850
25 "	125 to 250	3600	6 " 0 "	3600
75 "	125 " 250	2400	13 " 0 "	12000
150 "	125 " 250	2000	16 " 0 "	25500
300 "	125 " 250	1500	17 " 0 "	30000

Westinghouse-Parsons Turbine.—Fig. 104 is a longitudinal section through the Westinghouse-Parsons turbine, which illustrates the reaction type of machine.

In this turbine the reaction and impulse principles are combined by using a rotating drum on which there is a series of rings or blades, alternating with rings of stationary blades between them, which are attached to the casing. In Fig. 104 the moving blades are shown as rings encircling the enlarged shaft or drum of the turbine, while the stationary blades appear as teeth or projections between them. Both the stationary and movable rings are made up of crescent-shaped blades or vanes similar to those around the outer edge of the De Laval wheel. The relation borne by the moving blades to the stationary blades is shown more clearly in Fig. 100, where the direction of the steam is indicated by arrows.

Steam enters between the fixed blades and is deflected against the wheel blades beyond, forcing them forward by means of the impulse effect; then with but little loss in pressure it blows backward with nearly the same force on the farther side, and thus produces the reaction effect. The office of the stationary blades is simply that of deflectors to change the direction of the flow of steam coming from one row of moving blades and direct it toward the next.

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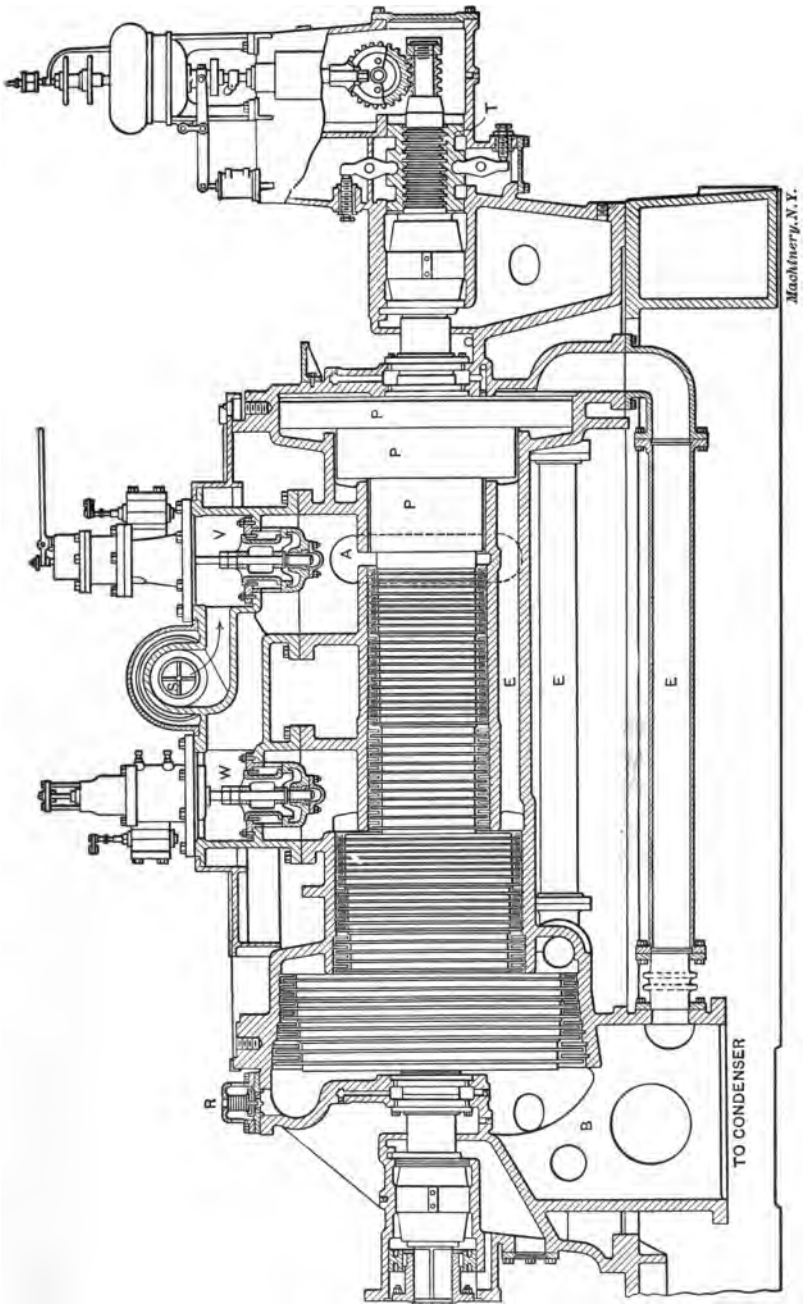

Fig. 104. Westinghouse-Parsons Reaction Turbine

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Steam is admitted to the chamber *A* through the governor valve, and passes through the blades as above described, to the exhaust chamber *B*. As it passes the different sets of blades, the volume of the passage is increased by enlarging the diameter of the drum and the height of the blades to correspond with the expansion of the steam.

In order to reduce the thrust which would result from the endwise pressure of the steam as it passes toward the exhaust, equalizing pistons *P*, *P*, *P* and passages *E*, *E*, *E* are provided. These pistons have the same areas as the enlarged portions of the shaft, and being exposed to the same pressures through the equalizing passages a balance is maintained.

As steam enters the turbine in puffs, and not in a continuous blast, speed regulation is accomplished by proportioning the duration of the puffs to the load. This is done by means of a small pilot valve actuated directly by the governor and which controls the steam supply through the main poppet admission valve. When the turbine is in operation the main poppet valve *V* is continually opening and closing at uniform intervals, but the periods during which the valve is allowed to remain open are proportioned to the load on the turbine. At light loads the valve opens for a very short period and remains closed during the greater part of the interval. As the load increases the period lengthens until finally, at about full load, the valve does not reach its seat at all and a continuous pressure is obtained in the high pressure end of the turbine. On the load becoming further increased an auxiliary or secondary valve *W* begins to open and admit steam to the annular space at the beginning of the intermediate drum of the rotor where the working steam areas are greater. This increases in proportion to the total power of the turbine. The operation of this secondary poppet valve is the same as that of the main admission valve, so that the governor automatically controls the power and speed of the turbine from no load to such overloads as are usually beyond the limits of generating apparatus built on normal ratings.

The governor is of the fly-ball type, the ball levers being mounted on knife edges instead of pins to secure sensitiveness.

Low Pressure Turbines.—These are designed to take the ex-

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haust steam from a reciprocating engine and expand it down to a pressure corresponding to about 28 inches vacuum.

Fig. 105 shows a turbine of this kind attached to a reciprocating engine.

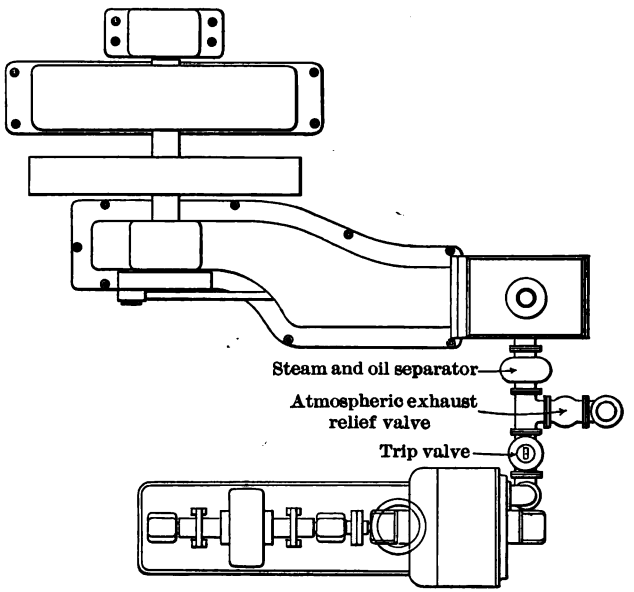

Fig. 105. Low-Pressure Turbine Attached to a Reciprocating Engine.

In general the steam economy of a simple engine and exhaust steam turbine together is always an improvement over that which would be obtained by compounding and adding a condenser. The plant capacity is increased from 50 to 100 per cent by exhausting through an exhaust turbine to a condenser from a non-condensing engine, while the increase by adding the turbine to a condensing outfit is from one and one half to three times that which could be secured from the condenser alone.

Mixed Pressure Turbines.—In cases where there is an ample supply of exhaust steam, or where the load carried by the turbine is in parallel with that on the engine, and fluctuates with it, the low-pressure type meets all requirements.

There are instances, however, where the available exhaust is

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not constant or where it is desired to obtain more power from the turbine than can be generated from low-pressure steam alone. Under these conditions, "mixed-pressure" machines are made use of, in which the low-pressure or exhaust steam is supplemented by high-pressure steam when required.

The method of supplying this will depend somewhat upon the conditions under which the turbine is to operate.

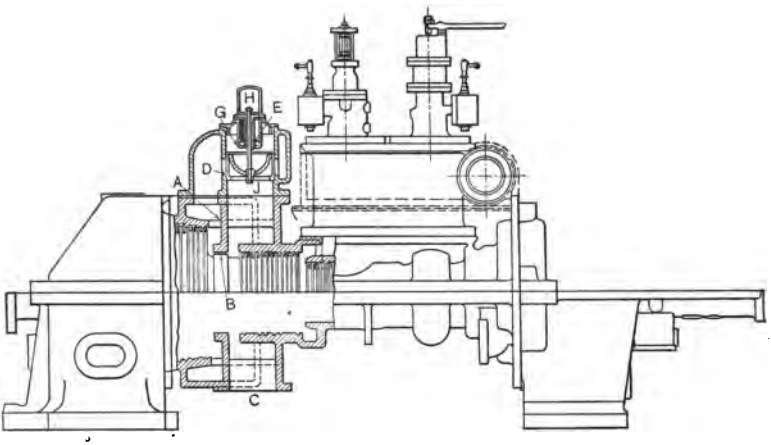

Fig. 106. Arrangement of Steam Turbine with Bleeder Connection and Constant Pressure Valve.

When high-pressure steam is only used occasionally, or in cases of emergency, it is usually admitted to the supply pipe through a reducing valve; but when depended upon regularly in considerable quantities it is best supplied through special nozzles similar to those employed for the low-pressure steam.

Bleeder Turbine.—This machine is designed especially for use in connection with steam heating plants.

Its principal advantages result from always using the steam economically before diverting it to the heating system, and at the same time being capable of automatically supplying large quantities of low pressure steam for central distribution or industrial purposes without affecting its electrical output.

Fig. 106 shows a Westinghouse-Parsons turbine of this type, the construction and operation of which are as follows:

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A division wall *A*, is introduced and strip packing provided at *B*, forming a labyrinth with no mechanical contact. Thus the steam leaving the intermediate stage may flow into the heating system through the nozzle *C* or else with a reduced heating demand, a part of the steam passes through the valve *D*, and performs work in the low pressure section. The automatic valve *D* is controlled by the desired pressure in the heating system and the balancing weight on the upper side *H*. The piston *E*, enclosed in an oil-filled chamber *G*, is provided to cushion any sudden movement due to pressure fluctuations. Thus if the pressure in the heating system is fixed at 3 lb. and the pressure at the end of the intermediate section tends to rise due to an increased turbine load, the valve opening will increase, and pass a greater portion of the steam to the low pressure stages. On the other hand, if the load decreases and the pressure at *J* falls, the valve opening will decrease until the pressure is again built up to 3 lb., the point determined upon.

Steam Turbine Economy

Mention has already been made that in general, the steam consumption of a turbine is practically the same as that of a reciprocating engine.

Tests seem to show that for sizes below 500 horse power, the turbine has proved rather more economical in the use of steam when compared with the single valve high speed engine, but that results are reversed with the four-valve compound engine of medium speed.

For larger sizes, ranging from 1,000 to 2,000 horse power, there seems to be but little difference in the economy of the two types of machines.

Comparing Engines and Turbines.—In making a comparison of the steam economy of reciprocating engines and turbines, the brake or delivered horse power should always be used as the indicated horse power of a turbine cannot be determined owing to the principle upon which it operates. When connected with an electric generator, the capacity may be expressed by the output in kilowatts.

Effect of Condensing.—The effect of condensing is much more marked in case of a turbine than with an engine, as will be made

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evident by reference to Fig. 129. If it were possible to plot the work done by a turbine in the form of a diagram, the ratio of the length to the area would be much greater than in the case of an engine, owing to the greater number of expansions. Therefore, any reduction in back pressure, which adds to the effective pressure throughout the stroke, will have a greater total effect. In other words, the addition to the diagram, represented by the enclosed area between the old and new back pressure lines, will be much greater in proportion in case of a long and narrow diagram than in one of the form shown in Fig. 129. The pronounced effect of this is evident when we consider that the best types of compound engines are usually limited to ten or twelve expansions, while in a turbine, one hundred or more expansions are easily obtained.

Load Variation.—A turbine has a much wider range of efficient operation under a variable load than a reciprocating engine. This adapts it especially to central station work, where the call for electricity varies greatly during the day.

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CHAPTER XIII.

AUXILIARY APPARATUS

Feed pumps.—Boilers are usually fed by small direct acting steam pumps, although power pumps driven by the engine are sometimes used. The former require a large amount of steam in proportion to the work done, but this is small in comparison with that generated by the boilers, and is offset to some extent by certain advantages which this type of pump has over the power pump.

The speed of the steam pump may be easily regulated to furnish the required amount of feed water under varying conditions, while with the power pump it is necessary to run it at a constant speed and allow the excess of water to flow back into the suction

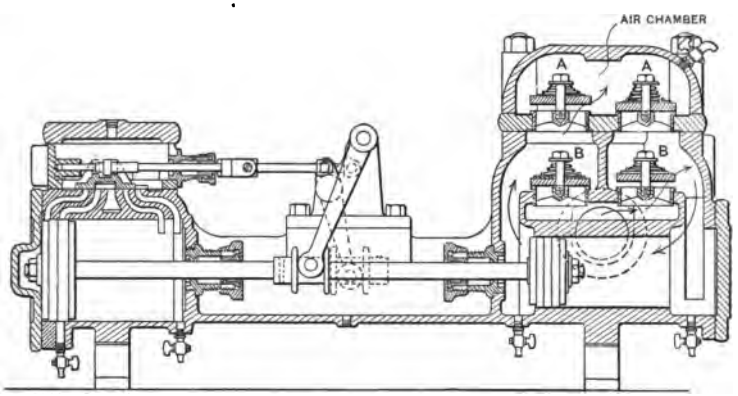

Fig. 107. Direct-Acting Piston Pump.

pipe through a relief valve. A section through a double acting piston pump is shown in Fig. 107. In construction it is like an ordinary slide valve steam engine with a water piston in place of the usual crosshead. The movement of the piston forces a part of the water in front of it through the upper valves *A A* into the air chamber, from which it flows through the discharge pipe to the boiler. On account of the partial vacuum formed back of the piston by its forward movement, water is drawn by suction into

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the cylinder through the valves *B B*, as indicated, ready for delivery on the return stroke. The valves are kept on their seats by light springs until the excess of pressure on the lower side is sufficient to lift them and allow the water to flow through.

Plunger Pumps.—The principal objection to the above type of pump is the difficulty experienced in keeping the piston tight and of detecting leaks when they occur. On this account the plunger pump is often preferred. Fig. 108 shows the water end of a double-acting single plunger pump. The advantage of this form over the piston is the ease with which it is kept tight under high

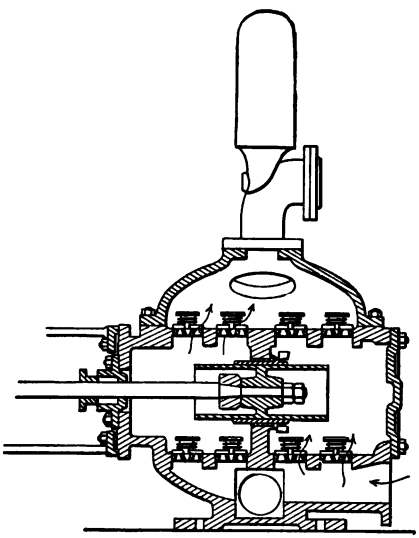

Fig. 108. Water End of Plunger Pump.

pressures, and with gritty liquids the wear is taken by the bushing, which is more easily and cheaply replaced than a cylinder lining. In order to examine or repack either of these forms it is necessary to dismantle the pump.

The plunger pump shown in Fig. 109 is so designed that any leakage may be at once detected and the glands repacked from the outside. This is indicated more clearly in Fig. 110. When two cylinders are placed side by side and discharge into a common delivery pipe, it is called a *duplex* pump.

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Capacity of Pumps.—Theoretically, the quantity of water delivered by a pump in a given time is equal to

Area of plunger × length of stroke × number of strokes.

Practically, there is a certain amount of leakage around the

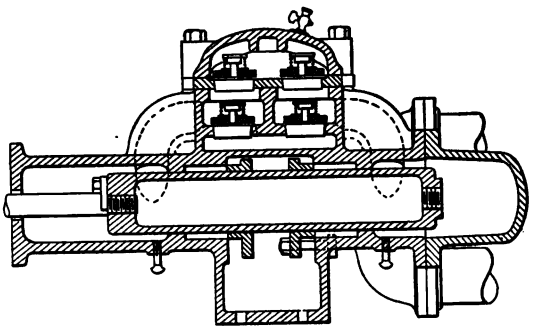

Fig. 109. Outside-Packed Plunger Pump.

plunger and through the valves called "slip." This brings the actual capacity down to about 80 per cent of the theoretical in well-made pumps of medium size working under ordinary conditions.

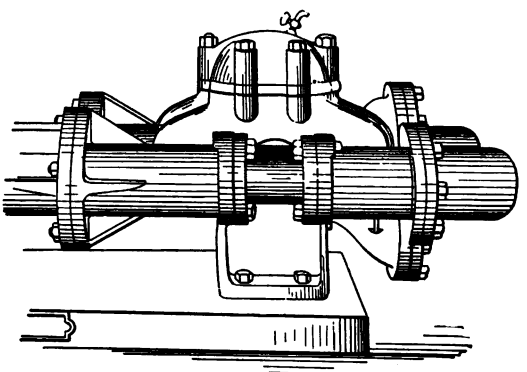

Fig. 110. Exterior View Showing Outside Glands.

Table LIV. gives the capacities of standard duplex feed pumps for various speeds. These figures are based on 30 pounds of water per hour per boiler horse power, and 80 per cent slip. The speed of a pump is often expressed in piston speed instead of

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strokes per minute. This should not exceed 100 feet per minute as a maximum, and 50 feet or less for continuous working is better.

The size of a pump is indicated by three dimensions, as 2 in. x 1⅛ in. x 2¾ in., in which the first is the diameter of the steam cylinder, the second the water cylinder, and the third the length of stroke. The proportions given in the table are for moderate to

TABLE LIV.

CAPACITIES OF BOILER FEED PUMPS.

Size of pump	30 strokes per minute		45 strokes per minute		60 strokes per minute		Approximate floor space	Steam pipe	Exhaust pipe	Suction pipe	Discharge pipe
	Lb. per hour	Horse power	Lb. per hour	Horse power	Lb. per hour	Horse power					
In. In. In.							In. In.	In.	In.	In.	In.
2 by 1⅛ by 2¾	198	7	297	10	396	13	5 by 20	⅜	½	1	¾
3 " 1¾ " 3	720	24	1080	36	1440	48	9 " 25	⅜	½	1¼	1
4¼ " 2¾ " 4	2316	77	3474	116	4632	154	12 " 33	½	¾	2	1½
5¼ " 3½ " 5	4800	160	7200	240	9600	320	14 " 38	¾	1½	2½	1½
6 " 4 " 6	7530	250	11295	376	15060	502	15 " 44	1	1½	3	2
7½ " 4½ " 6	9600	320	14400	480	19200	640	19 " 48	1½	2	4	3

high steam pressures. If it is desired to operate the pump at a pressure much less than 20 pounds, it should be provided with larger steam cylinders, depending upon the pressure at which it is desired to run it. In designing a boiler plant it is best to use two pumps of such capacity that one of them running at a speed of 50 or 60 strokes per minute will deliver the maximum quantity of water required under ordinary conditions. This makes easy work for the two running together and allows a reserve pump in case of a breakdown. The following data relating to water will be found useful in making calculations for the capacity of pumps:—

Cubic feet $\times$ 60 = pounds.

Pounds $\div$ 60 = cubic feet.

Gallons $\times$ 8.3 = pounds.

Pounds $\div$ 8.3 = gallons.

Cubic feet $\times$ 7.2 = gallons.

Gallons $\div$ 7.2 = cubic feet.

These figures will vary slightly for different temperatures, but are sufficiently accurate for ordinary use.

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Automatic Return Pumps.—In exhaust heating plants the condensation is commonly returned to the boilers by means of an automatic return pump. A typical outfit of this kind is shown in Fig. 111 and consists of a receiving tank mounted above a boiler feed pump as indicated. The bottom of the tank is connected with the suction of the pump and automatic operation is

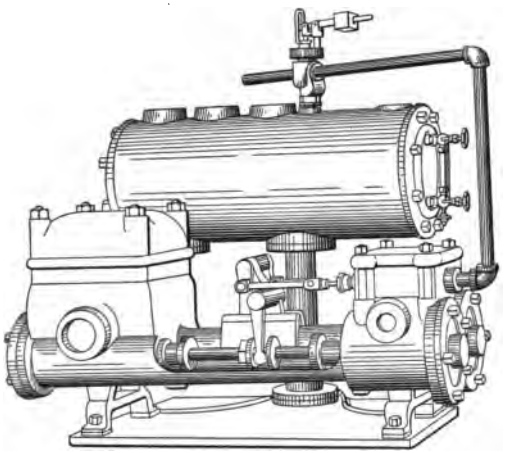

Fig. 111. Automatic Return Pump.

secured by means of an interior float connected with a balanced valve in the steam pipe leading to the pump. From this arrangement it is evident that changes in the water line within the tank will start and stop the pump as may be required to maintain a constant level.

Receiving tanks of this kind are usually vented to the atmosphere, and the condensation from different sources discharged into it by means of traps.

When the entire heating system is under one pressure, the trap may be omitted and the water of condensation be allowed to return directly to the receiver, which should of course be closed in this case.

When it is desired to seal the return mains the tank may be elevated to the required height and a "balance" pipe connected with the top for maintaining the same pressure in the upper part of the tank as exists in the heating system.

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In plants where practically the entire boiler output is returned in the form of condensation it is customary to connect the float with the steam valve, as shown in Fig. 111, and thus maintain a constant level in the tank, adding make-up water as needed by means of a hand valve.

In combined power and heating plants, where the condensation falls considerably short of the total supply required, it is better to run the feed pumps constantly, regulating the speed by hand, and attach the float to a valve in the fresh water supply pipe, thus automatically admitting the make-up water as needed.

TABLE LV.

CAPACITIES OF COMBINED PUMPS AND RECEIVERS.

Size of pump, inches			Over all dimensions, inches		
Steam cylinders	Water cylinders	Length of stroke	Length	Width	Height
3	2	3	37	22	34
4½	2¾	4	38	28	34
5½	3½	5	41	30	41
6	4	6	44	31	42
7½	4½	6	56	36	44

Table LV. gives the over-all dimensions of the Deane combined pump and receiver as shown in Fig. 111.

Although the returns from a heating system are often connected directly into the top of the receiver, it is better practice to join them below the water line. With this arrangement each return is sealed, and any slight difference in pressure is balanced by the water column in the vertical portion of the pipe.

Pump Governors.—These are often used in connection with a return pump in place of a receiving tank. They operate upon the same principle, the only difference being that the receiving chamber is made only large enough to accommodate the float which actuates the automatic valve, instead of serving also as an overflow for the condensation from the returns.

Fig. 112 shows the Mueller pump governor or regulator, the operation of which is evident from the cut. In case it is desired

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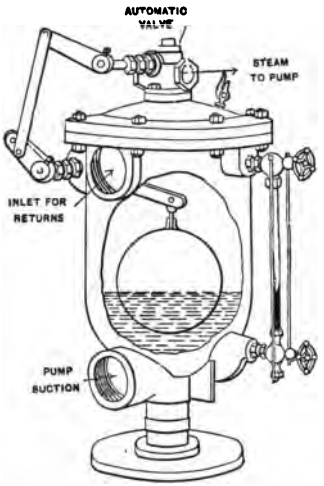

Fig. 112. Mueller Pump Governor.

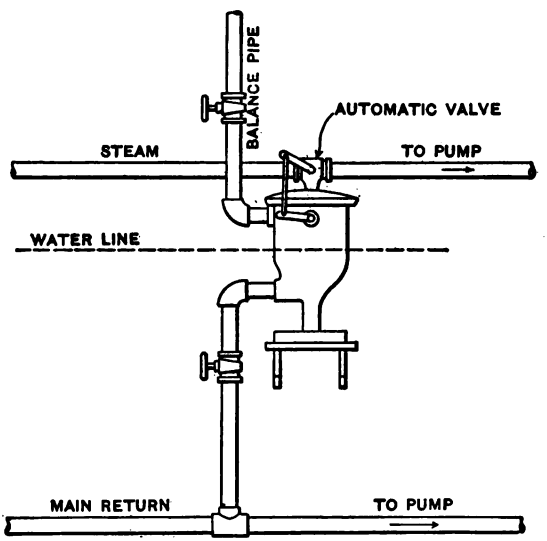

Fig. 113. Connections for Pump Governor.

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to seal the returns, the regulator may be raised to the required height upon a support, and the connections made as shown in Fig. 113. The upper opening in this arrangement is used for the connection of a balance or equalizing pipe instead of the inlet as before.

Traps

Traps are used for draining the water of condensation from steam pipes, coils, heaters, etc., without allowing the steam to

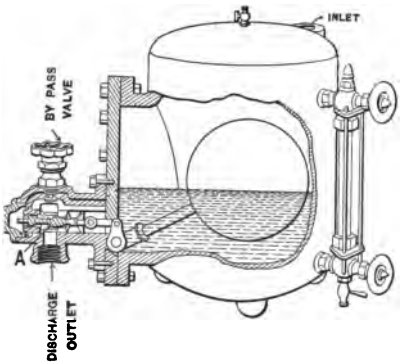

Fig. 114. Float Trap.

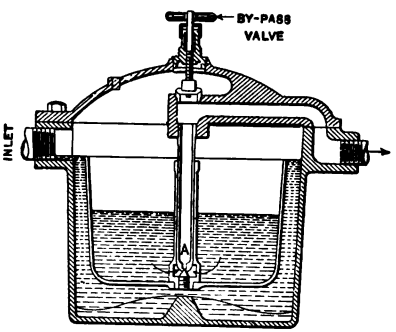

Fig. 115. Bucket Trap.

escape at the same time. There are a great many different forms, most of them, however, coming under the heads of *float, bucket,* and *expansion* traps. Fig. 114 shows the Curtis trap, a simple form of the first type. When the water line reaches a certain point, the float rises and opens the discharge valve *A*, which is placed near the bottom of the trap. As the water flows out, the float falls, and the valve closes before being uncovered, so that no steam is allowed to escape.

Bucket Traps.—A bucket trap is shown in Fig. 115. In this case the entering water fills the space around the bucket and floats it, thus closing the valve *A*. After the space becomes filled, the water overflows into the bucket causing it to sink. As the conical plug is attached to the bucket, the valve is opened, and the water flows through openings in the surrounding sleeve, as indicated by the arrows, and is forced upward through the inner tube by the steam pressure acting upon its surface. As soon as the bucket is partly emptied it again rises and closes the valve

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before the opening has become uncovered to admit the passage of steam. Both of the traps above described are provided with by-pass valves for blowing out.

Expansion Traps.—Fig. 116 shows the interior mechanism of the Heintz expansion trap. The valve closing the discharge open-

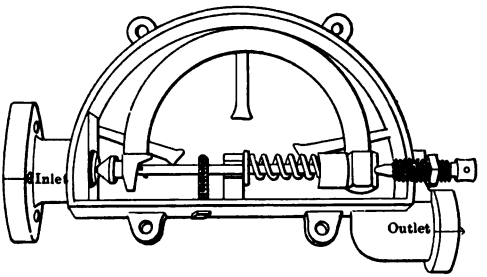

Fig. 116. Heintz Expansion Trap.

ing in this case is attached to the end of a curved tube partially filled with a volatile fluid. Water entering the trap at a temperature below 212 degrees will flow through without closing the

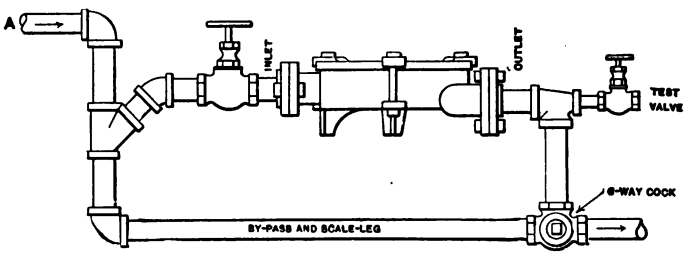

Fig. 117. Connections for Heintz Trap.

valve, but as soon as its temperature reaches this point, which is that of steam at atmospheric pressure, the vapor within the tube exerts a sufficient pressure to partially straighten it and thus completely closes the valve. The valve is in a wide open position when the temperature of the water is from 194 to 197 degrees, but is not completely closed until it reaches 212 degrees.

Fig. 117 shows the method of connecting up this type of trap. The pipe *A* should always be of sufficient length to allow the

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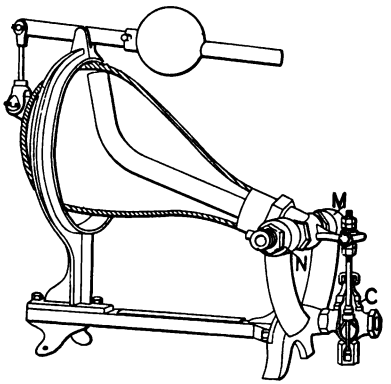

Fig. 118. Bundy Return Trap.

condensation to cool to a point below 212 degrees before reaching the trap. When two or more traps discharge into the same line, a check valve should always be placed in the outlet pipe.

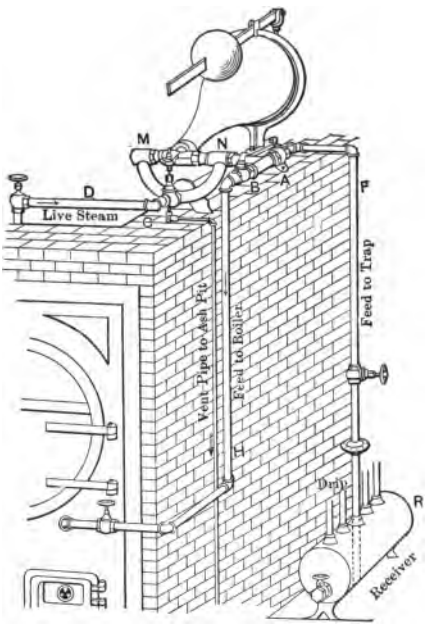

Fig. 119. Connections for Bundy Trap.

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Other expansion traps are operated by the expansion and contraction of tubes and rods made of iron and brass; the difference in the expansion of these two metals under the same conditions of temperature being made to open and close a valve in a similar manner to that in the trap just described.

Location of Traps.—Traps are usually placed below the pipe or a system of pipes to be drained, although they will operate in many cases if placed at a higher level. This is true because as soon as the pocket in the pipe leading to the trap is filled with water, it cuts off the supply of steam to the body of the trap, thus causing a vacuum which allows the water to be forced upward

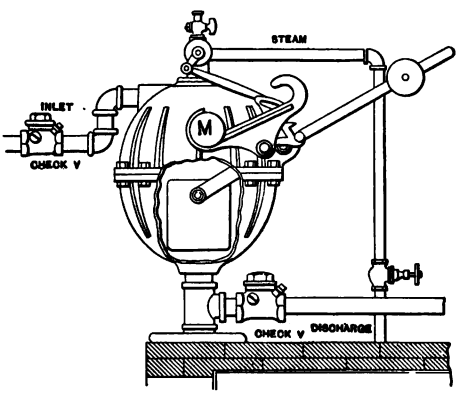

Fig. 120. Pratt Return Trap.

by the pressure in the system back of it. In case a trap discharges against atmospheric pressure, it will raise the water to a height of about 2 feet for each pound steam pressure within the trap.

All traps should be provided with a by-pass and cut-out valves; and where several traps discharge into a common main, it is well to place a check valve in the discharge pipe from each. Traps for low and high-pressure work are slightly different, and the pressure they are to operate under should always be mentioned when purchasing or specifying.

Return Traps.—The traps just described can only discharge their contents against a lower pressure.

A return trap is a device commonly used for receiving the condensation under a low pressure and discharging it back into

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the boiler against a much higher pressure. They are simple in construction and are often used instead of a pump in plants of small size. The Bundy return trap shown in Figs. 118 and 119 is a good illustration of this form of trap. It consists of a cast-iron bowl pivoted at *M* and *N*. There is an opening through *N* connecting with the inside of the bowl. The pipe *D* (Fig. 119), leading from the steam space of the boiler, connects through the valve *C* with an interior pipe opening near the top as shown in Fig. 118.

When in service, water enters the bowl from the receiver, until its weight overcomes that of the counterbalance and it falls to the bottom of the enclosing ring. This action opens the steam valve *C*, thus equalizing the pressures within the trap and boiler and allowing the water to flow from the bowl by gravity, due to its greater elevation. While the trap is discharging, communication with the feed pipe *F* is cut off by the check valve *A*, and when the bowl has emptied itself it is brought back to its original position by means of the counterweight, thus closing the steam valve *C*. As soon as the steam in the trap condenses, a vacuum is formed which draws up another supply of water from the receiver, and the operation is repeated. While filling, communication between the bowl and the boiler feed is closed by check valve *B*. To secure the best results the trap should be placed from 2 to 3 feet above the water line of the boiler.

In the Pratt return trap shown in Fig. 120 the equalizing valve is opened and closed by the action of a float and connecting levers.

There are several makes of return traps of different design, but all working upon practically the same principle, that is, the equalization of pressure in the trap and boiler.

Separators and Grease Extractors

Steam Separators.—If the water surface of a boiler is not sufficiently large for the free disengagement of the steam, a certain amount of moisture will be carried over in the form of spray; also, if steam is carried for some distance in a pipe, condensation will take place to a greater or lesser extent, depending upon the length of the pipe and its protection from the surrounding air. For the best results, steam should be delivered to an engine as dry as possible, and to provide for this, separators

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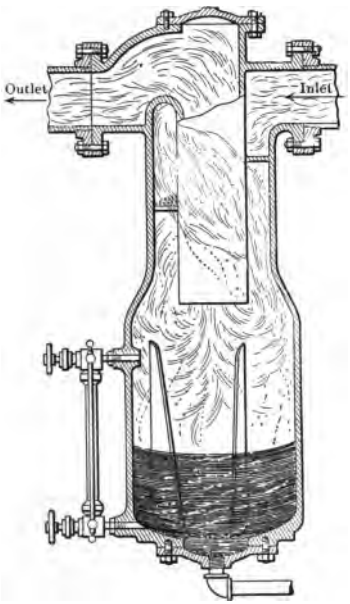

Fig. 121. Stratton Separator.

are commonly used. One form consists of a drum of considerable size, in which the current of steam comes to rest; the entrained water falls to the bottom and is drained away, while the dry steam flows from the top. In order to be effective the drum must be of ample size, or the water and steam will not be separated. There are several forms designed upon the principle that if the direction of a current of steam is suddenly changed, or if it flows downward and then upward, the water will be separated from the steam and fall to the bottom of the chamber in which the separation takes place. In the Stratton separator shown in Fig. 121 the steam enters at one side of a cylinder, flows downward with a rotary motion and then upward through a pipe at the center. Dry steam escapes from an opening near the top on the opposite side from which it enters, and the separated water is trapped away from the bottom. The Cochrane separator, Fig. 122, is of the baffle plate type. The connections for the entrance and exit of the steam project from each side of a spherical head. The baffle plate, which is cast as a part of the head, is ribbed or corrugated, and has ports at each side for the passage of steam. The area of these ports is made large to prevent loss of pressure by friction. Steam entering at the left-hand opening strikes the baffle plate and passes to the outlet chamber by means of the side ports shown in the cut. Separators are made in a great variety of forms, many of which are modifications of the general types above shown. In others the steam is given a whirling motion, and centrifugal force is relied upon to separate the

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water from the steam. They are made for use in both vertical and horizontal pipes.

Grease Extractors.—When the exhaust steam from engines and pumps is used for heating, provision must be made for removing the oil before the steam enters the heating system. This must be done for two reasons: First, a coating of oil or grease upon the inside of the radiators tends to reduce their efficiency as heating surface; and second, the presence of oil in the water of condensa-

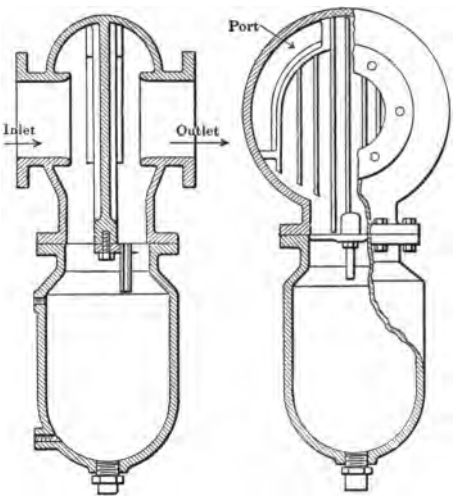

Fig. 122. Cochrane Separator.

tion returned to the boilers will give trouble in various ways. The latter is of the most importance and may cause foaming of the water, pitting of the plates, or increase the tendency to incrustation. There are various devices for removing oil from steam. Some of these consist of a series of baffle plates against which the particles of oil lodge as the steam passes among them. The Webster oil separator, shown in Fig. 123, is of this form. Other types depend upon the centrifugal action of the heavier particles of oil and water, and the steam is given a rotary motion which throws the oil to the outer walls of a collecting chamber, from which it trickles down to a receiver and is trapped away to the sewer. In other forms the steam is made to pass through a layer of excelsior, coke, or small pebbles.

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Sometimes, as an extra precaution, the water of condensation is again purified before it is returned to the boilers. A simple

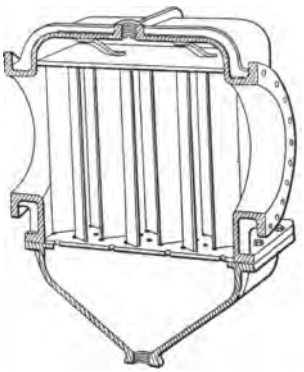

Fig. 123. Webster Oil Separator.

arrangement for this purpose is shown in Fig. 124. The settling

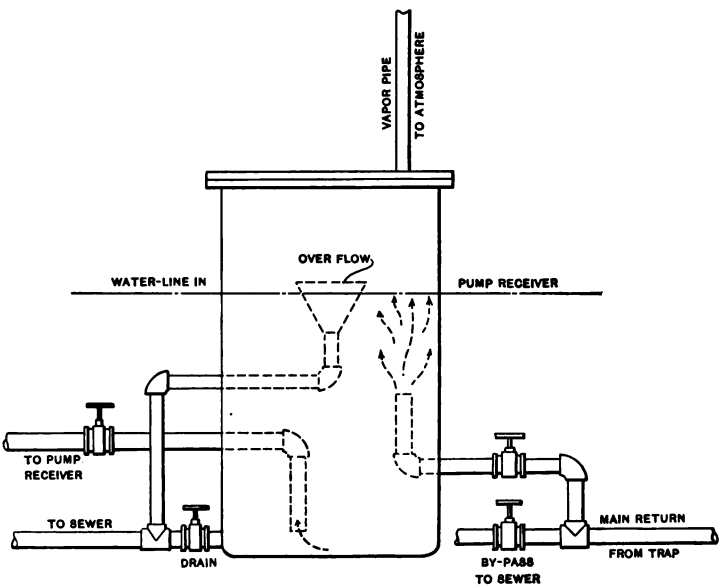

Fig. 124. Oil Separator and Purifer.

chamber may be made of cast or wrought iron, about 24 inches in diameter by 48 inches in height. The condensation from the

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heating system is trapped into the tank near the bottom, the discharge pipe ending in an elbow looking up. A second pipe connects the bottom of the tank with the automatic pump receiver, thus causing the water to stand at the same level in both. An overflow pipe ending in a tunnel inside the tank is provided as shown, the top of the tunnel being placed about an inch above the normal water line in the pump receiver. As the condensation is trapped into the tank the oil which is contains will rise to the surface, while the clearer water flows from the bottom of the tank into the pump receiver and is automatically returned to the boiler. By closing the discharge valve A occasionally, the water line is made to rise above the top of the tunnel and the oil overflows into the drain or sewer.

Feed Water Heating

Feed-Water Heaters.—Feed-water heaters are employed for two important reasons: First, if cold water is introduced into a boiler in any considerable quantity, the cooling effect will cause an unequal contraction of the plates, which is likely to produce leaks at the joints. It also tends to reduce the steam pressure, and necessitates more rapid firing for a time after feeding, unless the load upon the engines is practically constant, so that the feed valve can be set to supply a uniform quantity of water continuously.

Second, feed-water heaters can nearly always be arranged to utilize exhaust steam, or waste gases from the furnace, so that a substantial saving in fuel may be gained by their use.

Saving by Heating the Feed Water.—The percentage of saving in fuel by heating the feed water with waste products is expressed by the equation,

$$\frac{100\,(T_2 - T_1)}{L + (T_3 - T_1)}$$

in which

T_1 = initial temperature of the feed water entering the heater.
T_2 = final temperature of the feed water leaving the heater.
T_3 = temperature of the steam at boiler pressure.
L = latent heat of evaporation for steam at boiler pressure.

Example.—What will be the saving in fuel by using an exhaust

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steam feed water heater under the following conditions: Initial temperature of feed water 60°, final temperature 205°, and steam pressure on boilers 80 pounds gauge?

From Table I. we find the temperature of steam at this pressure to be 324°, and the latent heat of evaporation 886.

Substituting these values in the above equation, we have,

$$\frac{100\ (205—60)}{886+(324—60)}=\frac{14,500}{1,150}=12.6\%$$

This is true because the total heat required per pound of steam

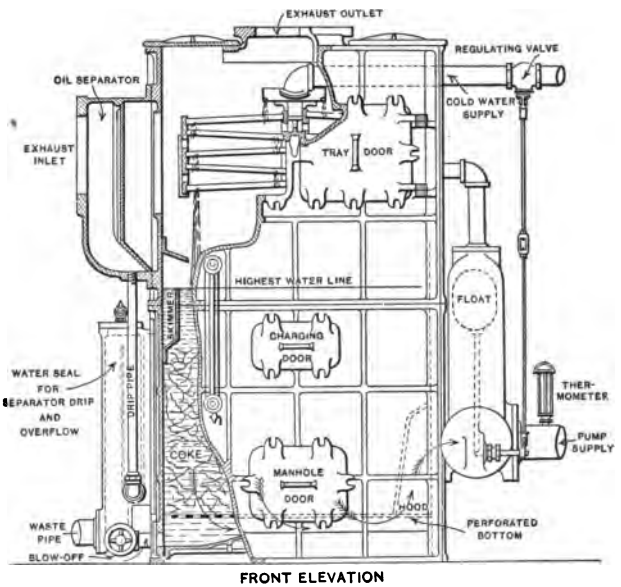

Fig. 125. Cochrane Open Feed-Water Heater.

is that necessary to raise the temperature of the feed water from 60° to 324°, which is 324 — 60 = 264 B. T. U., plus that required to evaporate it into steam at this temperature, which is 886 B. T. U., making a total of 264 + 886 = 1,150 B. T. U. The saving made by heating the feed water from 60° to 205° is 205 — 60 = 145 B. T. U. Therefore the gain in per cent is

$$\frac{145\times100}{1,150}=12.6$$

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With the average type of engine working under normal conditions only about 15 per cent of the heat stored in the steam leaving the boilers in transformed into work and lost in cylinder condensation. This leaves 85 per cent of the heat in the total amount of steam generated available for heating purposes, and

TABLE LVI.

DIMENSIONS OF COCHRANE FEED WATER-HEATERS.

Rated horse power	Diameter of exhaust connections	Height	Floor space
	In.	In.	In. In.
50	4	67	21 by 27
100	6	75	25 " 31
200	8	75	30 " 36
450	10	87	38 " 46
600	12	93	42 " 51

of this amount only a comparatively small proportion can be used for feed-water heating under ordinary conditions. In office buildings, factories, or other buildings having their own power plant, this remaining portion is usually employed for warming and ventilating purposes.

Heat Available for Feed-Water Heating.—The proportion of heat in the exhaust steam which can be used for feed-water heating may be easily determined as follows: The latent heat of evaporation for steam at atmospheric pressure is 966 B. T. U. That is, each pound of steam gives out 966 heat units when it is condensed. The quantity of heat required for raising the temperature of 1 pound of water from 50° to 210°, which may be taken as maximum requirements, is $210 - 50 = 160$ B. T. U. Therefore only $160 \div 966$, or approximately one sixth of the heat in each pound of exhaust steam can be utilized in warming the pound of feed water which is to replace it.

Heaters using exhaust or live steam are divided into two kinds, known as open and closed heaters. When the waste gases from the furnace are employed, the apparatus is commonly called an *economizer*.

Open Heaters usually consist of a chamber of cast iron or steel plate, into which the steam is admitted. Cold water is fed in at

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the top, and in most forms is made to fall in thin sheets by flowing over shallow trays placed one below the other. The intimate contact of the water with the steam heats it to a temperature very nearly that of the steam. When the water contains scale forming salts which are precipitated at temperatures below that to which the water is heated, the sediment is caught by the trays and can be removed from time to time as required. Heaters of this form are sometimes provided with filters which still further purify the water before it is pumped into the boilers.

As the condensed steam is mixed with the feed water, it becomes necessary to pass the exhaust through an efficient oil separator before it enters the heater.

Fig. 125 shows a section through the Cochrane feed-water heater and purifier, which is of the type just described. The cold water enters at the top and trickles downward over the intercepting trays to the receiver at the bottom.

When the water line in the receiver reaches a given point, a valve operated by a float closes the supply. Exhaust steam enters at the side through an oil separator and passes out at the top of

TABLE LVII.

DIMENSIONS OF GOUBERT VERTICAL HEATERS.

Rated horse power	Diameter of exhaust connections	Diameter of water connections	Height	Diameter of shell
	In.	In.	Ft. In.	In.
50	6	1¼	5 4	12
70	6	1½	6 6	12
100	6	1¾	8 2	12
200	8	2	8 4	16
300	10	2¼	8 9	21
400	10	2½	10 5	21
500	12	3	9 7	25
600	12	3	10 8	25
800	16	4	10 8	29
1000	16	4	12 4	29

the heater. The oil from the separator is trapped to the sewer. When used in connection with a heating system the returns are brought back and discharged into the receiver just below the water line. All drips from separators, steam jackets, etc., may be trapped into the receiver through a special opening.

Before returning to the boilers the water passes through a coke filter at the bottom of the receiving chamber as indicated.

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Table LVI. gives the overall dimensions of a few of the principal sizes of this heater.

Closed Heaters.—With a closed heater the feed water does not mingle with the steam, but passes through a series of brass or copper tubes which are surrounded with steam on the outside. The Goubert heater shown in Fig. 126 is a good illustration of this type. In this heater, the cold feed is supplied at the bottom and the hot water taken from the top. Closed heaters are made for use in both vertical and horizontal positions, the former, however, on account of the small floor space required, is usually preferable, unless it is desired to suspend it overhead. There are various forms of construction employed, some having straight tubes, while in others the heating surface is in the form of a coil. When straight tubes are used one end is made to slide in a stuffing box attached to the tube sheet in order to provide for the unequal expansion of the iron shell and the brass or copper tubes.

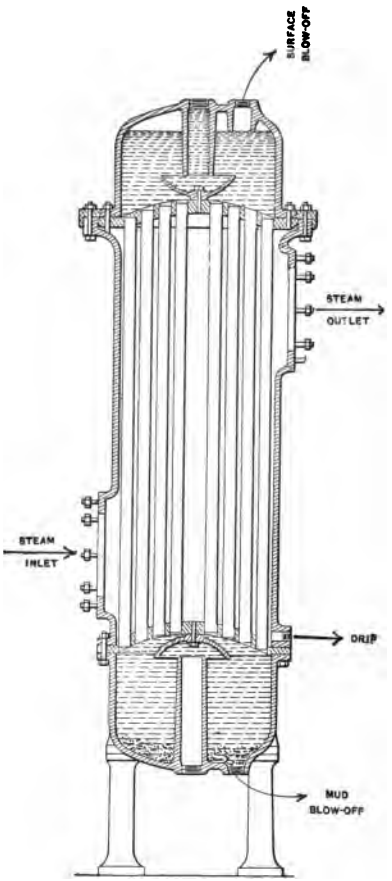

Fig. 126. Goubert Closed Heater.

The principal dimensions of some of the Goubert vertical heaters are given in Table LVII.

Heating Surface Required.—The required amount of heating surface depends upon the initial temperature of the feed water, the steam pressure within the heater, and the velocity with which the water flows through the tubes.

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With exhaust steam from non-condensing engines, it is customary to allow about 1 square foot of heating surface for each 90 pounds of water passed through the heater per hour. When the exhaust from condensing engines is used, the surface should be increased about 50 per cent, owing to the lower temperature of the steam. This calls for about 1/3 square foot of heating

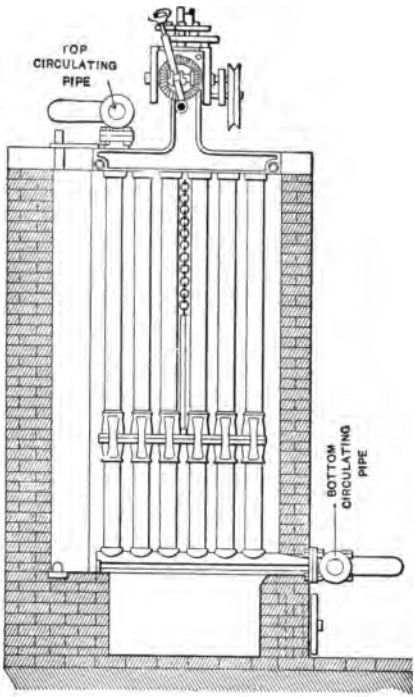

Fig. 127. Section of Green Economizer.

surface per boiler horse power for non-condensing engines and ½ square foot for condensing.

The temperature to which the feed water may be raised usually varies from 200° to 210° with exhaust steam from non-condensing engines, and from 110° to 120° in the case of condensing engines.

Heating Feed Water by Exhaust from Auxiliaries.—In many condensing plants the exhaust from the various pumps and other auxiliaries is sufficient to heat the feed water for the entire plant.

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When the exhaust from these sources is equal to about one sixth of the entire quantity of steam generated by the boilers, there will be nothing gained by placing a heater in the main exhaust pipe leading to the condenser; but if the exhaust from the auxiliaries is less than this amount, the feed water may be first passed through a heater supplied with steam at condenser pressure, and then through a second heater taking the pump exhaust at atmos-

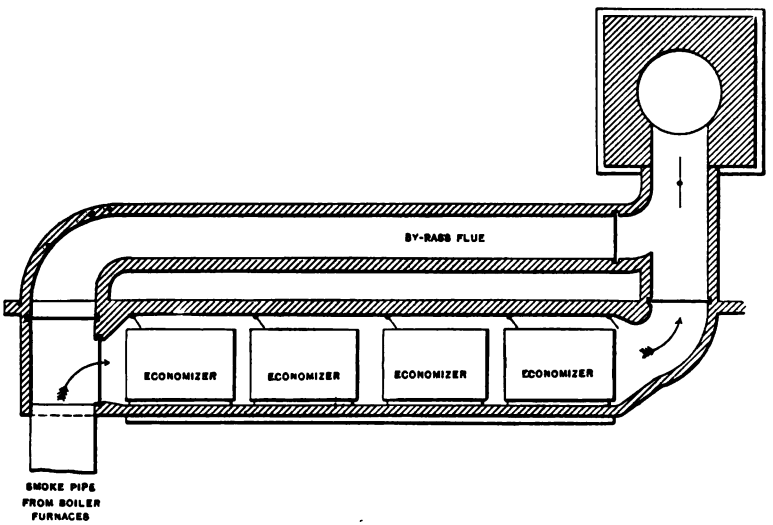

Fig. 128. Layout for Economizers.

pheric pressure, which will heat the water up to a temperature of about 205°.

Economizers

Construction.—Economizers are commonly made up of rows of vertical cast iron pipes about 4 inches in diameter and 9 feet high, connected at top and bottom by headers, each header, in turn, being connected with pipes running lengthwise, one at the top and one at the bottom, outside the brick chamber enclosing the apparatus. Each tube is provided with a geared scraper which travels up and down continuously for the purpose of removing the soot as it collects.

A single section of the Green economizer with its surrounding brickwork is shown in Fig. 127. The mechanism for operating

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the scrapers is seen at the top and may be driven by a belt from some convenient shaft or by an independent engine or motor. The feed water is forced into the economizer by the boiler pump at the lower branch, nearest the point of exit of the gases, and leaves through the upper branch pipe at the other end.

The series of pipes and headers are enclosed in a brick chamber through which the furnace gases are made to pass on their way to the chimney. The space below the lower headers is for the accumulation of soot removed by the scrapers, and must be cleared at frequent intervals.

Location of Economizer.—There are various ways of locating the economizer with relation to the boilers and chimney flues, depending upon the conditions to be met in each particular case. Fig. 128 shows a typical layout. Dampers and flues are provided so that the gases may be passed either through the economizer or turned directly into the chimney, in case it is desired to clean or repair the economizer at any time.

Saving Effected by Economizers.—The percentage of saving by the use of an economizer will depend on circumstances, but can be determined approximately by the general formula given for feed water heaters. The average of nine tests, reported by Mr. William R. Roney in a paper read before the American Society of Mechanical Engineers showed the rise in temperature of the feed water in passing through the economizer to be 140°, with a corresponding drop of 257° in the temperature of the furnace gases. The average saving in fuel for the same tests was 13.8 per cent. These results will of course vary somewhat in different cases.

While economizers are commonly used for the purpose of reducing the cost of fuel, they are often installed in connection with existing boiler plants for increasing the capacity. The addition of an economizer is, in effect, increasing the heating surface of the boilers, although the additional surface cannot be rated on the same basis per square foot, owing to the reduced temperature of the gases to which it is exposed.

The percentage of saving in fuel by the use of an economizer may be taken as approximately the same as the increase in boiler capacity, should the same amount of fuel be burned as before the economizer was installed.

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Area of Heating Surface.—It is customary under ordinary conditions to provide from 4½ to 5 square feet of heating surface in the economizer for each boiler horse power. Table LVIII. will be found useful in determining the approximate space required

TABLE LVIII.

APPROXIMATE SPACE REQUIRED FOR ECONOMIZERS.

Number of tubes wide	Number of rows	Total length of economizer	Height over		Dimensions inside walls			Area between tubes			External heating surface
			Gearing	Section	Without side dampers	With one side damper	With two side dampers	Without side dampers	With one side damper	With two side dampers	
		Ft. In.	Ft. In.	Ft. In.	Ft. In.	Ft. In.	Ft. In.				
4	8	5 10	13 2½	10 2½	3 4	4 1	4 10	16.6	23.85	31.10	384
6	8	4 10	" "	" "	4 8	5 5	6 2	21.85	29.10	36.35	576
8	12	7 3	" "	" "	6 0	6 9	7 6	27.00	34.25	41.5	1152
10	16	9 8	" "	" "	7 4	8 1	8 10	32.25	39.50	46.75	1920

for an economizer. This table gives only short lengths made up of 8 to 16 rows of tubes. In practice they are often made 60 or more rows deep. Any combination to give the required heating surface, flue area, etc., can be easily made up from the data given.

When it is desired to keep the resistance through the economizer as low as possible, the sections may be arranged in two groups with a by-pass flue between them, instead of placing them in a single row as in Fig. 128.

Condensers

Object of a Condenser.—The expansion of steam in the cylinder of an engine causes its pressure to gradually fall until it finally reaches a point where it cannot be profitably used for driving the piston. At this stage some means must be provided for disposing of the vapor, in order to prevent back pressure during the return stroke of the piston. If the atmosphere did not exert a pressure of its own, the steam could be disposed of at once by exhausting into the open air. But there is in reality a

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pressure of nearly 15 pounds per square inch due to the weight of the atmosphere, from which it follows that the steam from a non-condensing engine cannot be expanded below this pressure to any advantage, as it must eventually be exhausted against atmospheric pressure on the return stroke.

If by any means this back pressure can be removed it is evident that the engine will not only be aided by relieving the resistance upon the exhaust side of the piston, but the steam can be expanded in the cylinder nearly to absolute zero pressure, thus utilizing practically its full expansive power.

If the steam is discharged into a closed chamber in such a manner that it comes in contact with a spray of cold water or with a series of tubes through which cold water is circulated, it will be deprived of nearly all of its latent heat and will condense. In either case the act of condensation is practically instantaneous. As the water of condensation is only about 1/1600 of its original volume, it follows that the remainder of the space is void and that no pressure exists. If the expanded steam from an engine is conducted into this empty space, it will meet with no resistance and the limit of its usefulness will be reached. An arrangement of this kind in practical form is called a *condenser*. The cold water that produces the condensation is the *injection* or cooling water, and the heated water on leaving the condenser is called the *discharge* water.

Measurement of Vacuum.—The state of vacuum in a condenser is usually expressed in inches of mercury. For example, if the chamber of a condenser be connected by a rubber tube with the space above the mercury in a barometer, and the mercury column should fall to 26 inches, the condenser would be said to have 26 inches of vacuum. If a barometer stood at 30 inches before the rubber tube was attached, that is, when the space above the mercury was a vacuum, it would indicate that there was a pressure in the condenser equal to $30 - 26 = 4$ inches of mercury, or $4 \times .49 = 1.96$ pounds per square inch, which corresponds to a temperature of about 126°, and if the condenser were absolutely air-tight, would indicate that the water and vapor within were at that temperature. In order that the air may not accumulate in the condenser and destroy the vacuum, it is withdrawn, together with the condensed steam, by means of a pump.

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Weight of Injection Water.—The quantity of water required to completely condense the steam from an engine depends on two conditions: the pressure and weight of steam and the temperature of the injection water. With 26 inches of vacuum and injection water not above 70°, the required amount of injection water will be approximately from twenty-five to thirty times the weight of steam condensed. If the temperature of the injection water is lower, a less quantity is required, and if higher, the amount must be increased; for example, at 80°, thirty-five times, and at 90°, fifty-two times the weight of steam is required. The exact amount in each special case can be computed, theoretically, by the following formula:

$$Q = \frac{S—(D—32)}{D—I}$$

in which

Q = weight of injection water per pound of steam.

S = total heat of steam above 32° at exhaust or terminal pressure.

I = temperature of injection water.

D = temperature of discharge water.

The use of the formula will be made clear by applying it to a practical example.

Example.—A 100 horse power engine, requiring 20 lb. of steam per horse power, has a terminal pressure of 10 lb. absolute. The temperature of the injection water is 70° and we will assume a temperature of 120° for the discharge water, which corresponds approximately to 26 in. of vacuum. What will be the weight of injection water required?

From the above conditions we have

$$S = 1{,}140.9$$
$$I = 70$$
$$D = 120$$

Substituting in the formula, we have

$$Q = \frac{1{,}140.9—(120—32)}{120—70} = 21 +$$

The weight of steam required to supply the engine is 100 × 20

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= 2,000 lb. per hour. Therefore, neglecting cylinder losses, the weight of injection water is

$$2{,}000 \times 21 = 42{,}000 \text{ lb.}$$

The reason for the above formula is evident. One pound of steam at 10 lb. pressure contains 1,140.9 B. T. U. above a temperature of 32°. That is, if the steam were condensed and cooled to 32°, 1,140.9 B. T. U. would be given off. But in the example it is only cooled to 120°. One B. T. U. will raise the temperature of 1 lb. of water 1°, or, in other words, 1 B. T. U. will remain in the condensed steam for each degree that its final temperature is above 32°.

Therefore, the heat given out by condensing 1 lb. of steam at 10 lb. pressure and cooling it to 120° is 1,140.9 — (120 — 32) = 1,052.9 B. T. U. The heat absorbed by 1 lb. of water in raising it from a temperature of 70° to 120° is 120 — 70 = 50 B. T. U.

Therefore, 1,052.9 ÷ 50 = 21 +, the pounds of water which will be required to absorb the heat given off by the condensation and cooling of 1 lb. of steam under the conditions stated in the example. The matter of subtracting 32 from the temperature *D* affects the result but little and is often omitted in practice, the formula being written

$$Q = \frac{S-D}{D-I}$$

but for purposes of demonstration it is necessary to include it.

TABLE LIX.

TERMINAL PRESSURES IN STEAM ENGINE CYLINDERS.

Clearance	0	1.75%	3%	5%	7%	9%
Cut-off						
1/10	0.1	0.115	0.126	0.143	0.159	0.174
1/6	0.167	0.101	0.191	0.206	0.221	0.235
1/4	0.25	0.263	0.272	0.286	0.299	0.312
1/3	0.333	0.344	0.353	0.365	0.377	0.388
3/8	0.4	0.410	0.417	0.428	0.439	0.449
1/2	0.5	0.509	0.514	0.524	0.533	0.541
3/4	0.75	0.754	0.755	0.762	0.766	0.771
9/10	0.9	0.902	0.903	0.905	0.906	0.908

To find the (theoretical) absolute terminal pressure, multiply the absolute initial pressure by the number opposite the given cut-off and in the column corresponding to the percentage of clearance.

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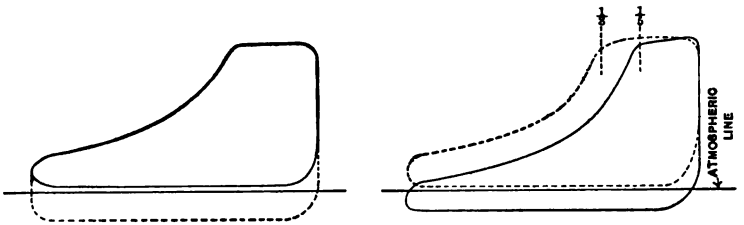
In practice, however, the final temperature of the cooling water is considerably below that of the condensed steam, so that more water must be provided than is given by the formula. Where the exact conditions are not known it is customary to allow about 35 pounds of water per pound of steam when the condenser is attached to a reciprocating engine and twice that amount when used with a turbine.

Table LIX. will be found useful for determining the terminal pressure under different conditions.



Figs. 129 and 130. Showing Gain from Condensing.

Advantages of Running Condensing

The advantages gained by using a condenser are an increase in power of an engine and also a greater economy in the amount of steam per horse power developed. The power exerted by a steam engine during a single stroke is due directly to the difference between the pressures acting on the opposite sides of the piston. A vacuum does not in itself give power, but it removes the resistance from the exhaust side of the piston, and therefore adds just so much to the steam side.

Gain in Power by Condensing.—The value of a vacuum of 26 inches of mercury to an engine, making due allowance for the cost of production, may be approximated by considering it equivalent to a net gain of about 12 pounds average pressure per square inch of piston area. This is shown graphically in Fig. 129, in which the full line represents the indicator card when exhausting into the atmosphere. If the exhaust be connected with a condenser and the points of cut-off and release remain the same, the back-pressure line will drop to the position shown by the dotted line, and the difference between the two back-pressure lines will be added directly to the mean effective pressure, because it is a

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constant effective pressure extending throughout the entire stroke. From this it is evident that the power gained bears practically the same ratio to the power developed by the engine when non-condensing, as 12 pounds does to the mean effective pressure when running under the same conditions. So if the mean effective pressure of a non-condensing engine is known, a close approximation of the gain in power that will be derived by the use of a vacuum is easily determined.

The formula for computing the indicated horse power of an engine has already been given as

$$\frac{P \times L \times A \times N}{33{,}000}$$

in which

P = mean effective pressure in pounds per square inch acting on the piston (M. E. P.)

L = length of stroke in feet.

A = area of piston in square inches.

N = number of single strokes per minute.

By substituting 12 for P in this formula, it will give the horse power gained by a vacuum of 26 inches when applied to a non-condensing engine.

Example.—An 18″ × 36″ non-condensing engine runs at a speed of 200 revolutions per minute and cuts off at ¼ stroke; boiler pressure 95 lb. gauge. What gain in horse power will be obtained by running condensing with a vacuum of 26 in., and what will be the final horse power?

Referring to Table XLVI., page 163, and neglecting clearance, we have, M. E. P. = $[(95 + 15) \times 0.60] - 17 = 49$, and the I. H. P. when non-condensing is

$$\frac{49 \times 3 \times 254 \times 400}{33{,}000} = 452$$

When running condensing the I. H. P. is found in the same way by taking a back pressure of 3 lb. instead of 17, and we have

$$\text{I. H. P.} = \frac{63 \times 3 \times 254 \times 400}{33{,}000} = 582$$

as the final H. P. with a gain of $582 - 452 = 130$ H. P.

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The gain in power could also have been found approximately by substituting 12 for 49 in the first equation.

Gain in Economy by Condensing.—The desirability of installing a condenser in any particular case will depend upon various conditions, among which are the size and water rate of the engine, cost of injection water, cost of fuel, and also to what extent the exhaust steam can be used for heating purposes.

The saving in fuel by the use of a condenser, when the power remains the same, is illustrated in Fig. 130. These two diagrams were taken from the same engine. The first, shown by the dotted lines, was obtained when running non-condensing, and the second when running condensing. The areas of the two cards are equal, so that the work done per stroke is the same in each case.

The cut-off, however, occurs at approximately 1/3 stroke in the first and 1/5 in the second, so that theoretically, the difference between 1/3 and 1/5 of a cylinder of steam is saved at each stroke. Actually it will be somewhat less than this, as the cylinder condensation will increase with the ratio of expansion.

Under favorable conditions in actual practice, it is customary to expect a saving in fuel of about 28 to 30 per cent in the case of simple engines, and 20 to 22 per cent for compound engines by running condensing. In the case of office buildings, where the greater part, if not all, of the exhaust steam can be used during the winter months for heating, and where the injection water must be purchased at city rates, there is nothing to be gained by condensing, and in many cases it would probably result in a loss. On the other hand, in manufacturing plants where water for condensing purposes can be had free of charge, or in large electric power plants where the exhaust steam is wasted, condensers are very generally employed.

Example.—Taking the same engine as in the previous example, what percentage of saving could be made by using a condenser and shortening the cut-off to keep the power the same?

The first step is to find the new point of cut-off which will be required to maintain the same power of engine. This may be obtained algebraically by the formula already given for the M. E. P.

In this case the initial and mean effective pressures remain the

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same, the only change being in the back pressure, so that we may write

$$[(95 + 15) \times R] - 3 = 49,$$

and solve for the ratio R, which gives us

$$(110\,R) - 3 = 49,$$

or

$$R = \frac{49 + 3}{110} = 0.47$$

Looking in Table XLVI., we find that the nearest ratio, 0.46, corresponds to a cut-off of 1/6, which we may take in this case.

The saving in steam per stroke will then be

$$\tfrac{1}{4} - \tfrac{1}{6} \div \tfrac{1}{4} = \frac{0.25 - 0.166}{0.25} = 0\ 33 \text{ or } 33\%$$

While a vacuum approaching 26 inches is often obtained in first-class plants, the average more commonly runs from 20 to 24 inches, when the condenser is used with a reciprocating engine. Under these conditions the net gain will be more nearly approximated by assuming an increase of 10 pounds in the M. E. P. instead of 12, as used in the preceding illustrations. In case of a steam turbine, a higher vacuum is essential for the best efficiency, and vacuums of 26 to 28 inches are commonly maintained in this class of work.

Types of Condensers

The condensers in most common use are divided into three general classes, known as surface, jet and siphon condensers. In the former the exhaust steam enters a chamber filled with small tubes through which water is made to circulate, and condensation is brought about by the steam coming in contact with the colder surface of the tubes. The cooling water is forced through the tubes by means of a circulating pump, and an air pump, so called, is provided for removing the condensed steam and air from the vacuum chamber. The surface condenser is employed almost exclusively for marine work, while both the surface and jet condensers are used on land. It is heavier and more bulky than the jet condenser for any given capacity, but it may be used with any kind of water, which is not the case with the jet condenser if the discharge water is to be fed into the boilers. The

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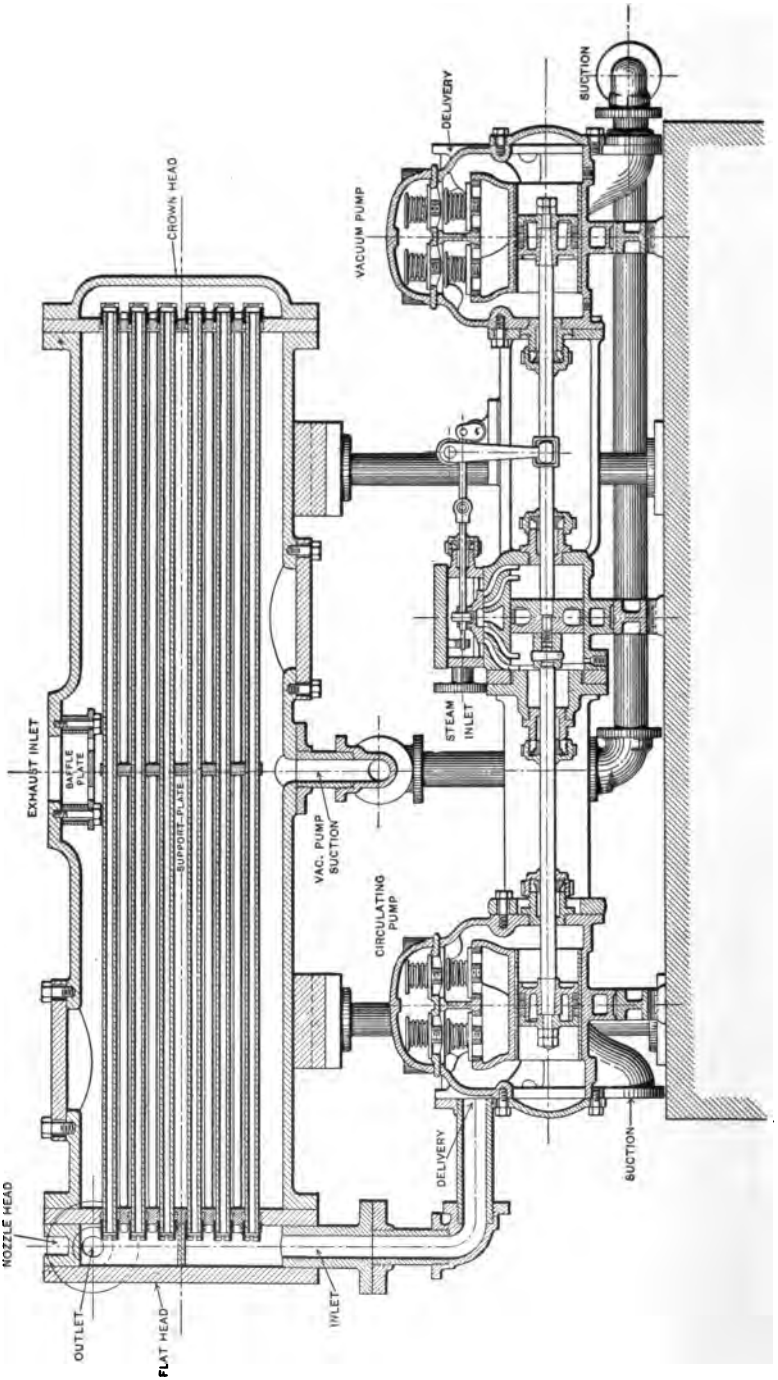

Fig. 131. Section of Surface Condenser with Combined Air and Circulating Pump.

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condensed steam from a surface condenser may be returned to the boilers, provided from 10 to 12 per cent of its weight of fresh water is added. Otherwise the effect of the distilled water is to corrode the boiler plates. The oil in the exhaust steam must be separated by means of an oil separator before it enters the condenser. This type of condenser is effective when the only available water supply contains solid matter, salts or acids which would be injurious to the boilers. The same water is used over and over again, except for the dilution with fresh water as above stated, so that the steam circuit is practically a closed one.

In the jet condenser the steam and injection water mingle together in the condensing chamber, and the condensed steam, air, and discharge water are removed by means of the same pump.

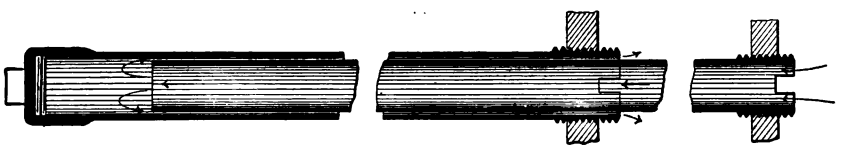

Fig. 132. Tube of Wheeler Condenser.

The siphon or barometric condenser is especially adapted to cases where the cooling water is of good quality and suitable for boiler feeding. It may also be used with any quality of water, if the condensation of steam is the only object sought and the cooling water is not to be used in connection with the boilers. When the right elevations are available this type of condenser operates without the use of pumps or valves and requires practically no adjustment.

Surface Condensers.—A typical form of surface condenser is shown in section in Fig. 131. The condensing chamber in this case is made of cast iron and is supported above the pumps. The water or circulating pump is shown at the left and the air or vacuum pump at the right. The water, air, and steam pistons are attached to a common rod, with the steam cylinder at the center.

In operation the exhaust steam enters the top of the chamber where it strikes a baffle plate for the purpose of distributing it over the cooling surfaces and also for preventing any cutting of the tubes near the inlet.

The steam, after being condensed, passes with the air and other

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non-condensable vapors to the vacuum pump as shown. The cooling water is delivered by the circulating pump into the lower part of the condensing chamber at one end. From here it passes successively through the lower and upper groups of the tubes, in reverse directions, and is discharged from the same end of the condensing chamber near the top.

In some makes a double tube of the form shown in Fig. 132 is made use of, instead of passing the water through two groups of tubes, in opposite directions, as in Fig. 131. The circulation through a tube of this kind is clearly indicated by the arrows.

Area of Cooling Surface.—The required cooling surface of the tubes may be computed by the equation

$$S = \frac{WL}{180\,(T-t)}$$

in which

S = the required cooling surface in square feet for brass tubes.
W = weight of steam to be condensed per hour.
T = temperature of steam at condenser pressure.
t = the average temperature of the circulating water, which is approximately the arithmetical mean of the initial and final temperatures.
L = the latent heat of steam at temperature T.

TABLE LX.

OVER-ALL DIMENSIONS OF WHEELER SURFACE CONDENSER.

Square feet condensing surface	Pump cylinders Size, inches	Length Ft. In.	Width Ft. In.	Height Ft. In.
150	5 and 7 by 6	6 2	1 9	4 0
200	5 " 7 " 6	6 9	1 9	4 0
300	6 " 7 " 8	6 8	2 1	6 0
600	7½ " 10 " 10	9 4	1 9	6 0
900	10 " 12 " 12	9 8	2 4	6 11
1200	10 " 12 " 14	9 9	2 5	7 4
1500	12 " 16 " 16	12 0	2 10	8 4
2000	12 " 16 " 16	12 0	3 2	8 8
2400	12 " 16 " 18	12 5	4 2	8 9

Example.—What area of cooling surface is required in a condenser to be used with a 100 horse power engine using 25 lb. of steam per horse power per hour? Assume for ordinary con-

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ditions that the pressure in the condenser is 3 lb. absolute, and that the initial temperature of the circulating water is 70° in summer time, with a final temperature of 110°.

We have from the foregoing formula, neglecting cylinder losses,

$$W = 100 \times 25 = 2{,}500$$
$$T = 141$$
$$t = \frac{70 + 110}{2} = 90$$
$$L = 1{,}015$$

which, substituted, in the formula, gives

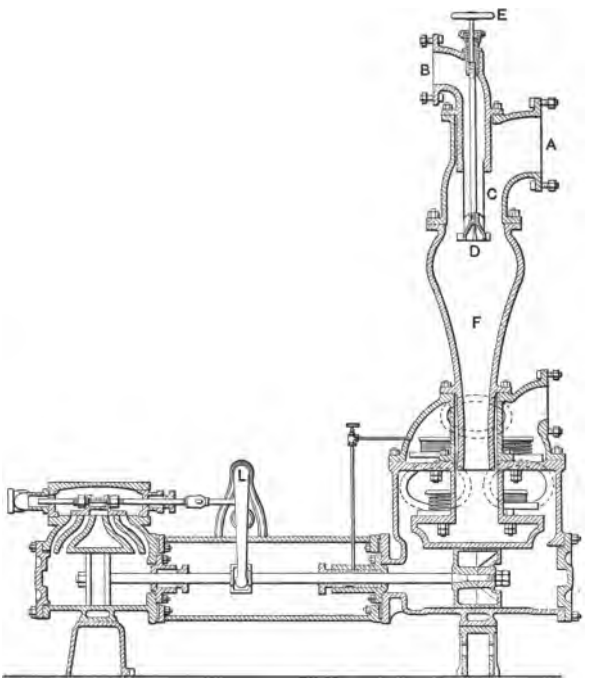

Fig. 133. Jet Condenser.

$$S = \frac{2{,}500 \times 1{,}015}{180\ (141 - 90)} = 276$$

or, 276 ÷ 100 = 27 sq. ft. of cooling surface per H. P.

Table LX. gives the over-all dimensions of a few standard sizes of the Wheeler surface condenser. These, however, are

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made in many different sizes and with varying proportions to meet all conditions.

Jet Condensers.—A common form of jet condenser is shown in Fig. 133. As already stated, the injection water acting directly on the steam will produce a greater drop in pressure with less water than a surface condenser, and the bulk and weight of the condenser itself is also less. When the condensed steam is to be pumped back into the boilers, the injection water goes with it, so that in

TABLE LXI.

DIMENSIONS AND FLOOR AREA OF KNOWLES JET CONDENSERS.

Pounds of steam condensed per hour	Diameter of steam cylinder	Diameter of air cylinder	Stroke	Floor space	
				Width	Length
	In.	In.	In.	Ft. In.	Ft. In.
1560	5½	8	7	2 5	4 0
2240	6	9	10	2 10	4 10
3340	6	10	12	3 4	5 7
4800	8	12	12	3 4	5 7
6400	8	14	12	3 10	6 0
6540	8	14	16	4 0	6 7
8540	10	16	16	4 6	7 0
10800	12	18	18	4 6	7 10
13340	12	20	24	5 11	9 8
16140	14	22	24	5 10	9 9

this case the water supply must be of a quality which is not injurious to the boiler plates.

Referring to Fig. 133, the action is as follows: The exhaust steam from the engine enters the condensing chamber at *A;* the injection water enters at *B*; *C* is the spray pipe and has at its lower extremity a number of vertical slits through which the water passes and becomes spread into thin sheets. The spray cone *D* breaks the water passing over it into a fine spray and thus causes a rapid and thorough mixture of the steam and water. The spray cone is adjusted to give the proper amount of water by means of a stem passing through the top of the condenser.

The injection water and condensed steam fall together through the opening *F* into the pump and are discharged either into the sewer or into a hot-well when the discharge water is to be used for feeding the boilers. The condensing chamber is ordinarily made from 1/3 to ½ the volume of the engine cylinder with which it is to be used.

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The injection water may be raised from a tank or other supply by the vacuum action of the condenser, provided the elevation is not more than 20 feet. If more than this, some form of pump must be used. The apparatus is put in operation by first starting the pump to produce a partial vacuum in the condensing chamber. This causes the injection water to enter through the pipe attached at *B*. The main engine may now be started, and as the exhaust enters through *A* it meets the spray of cool water and is rapidly condensed, thus maintaining a constant vacuum.

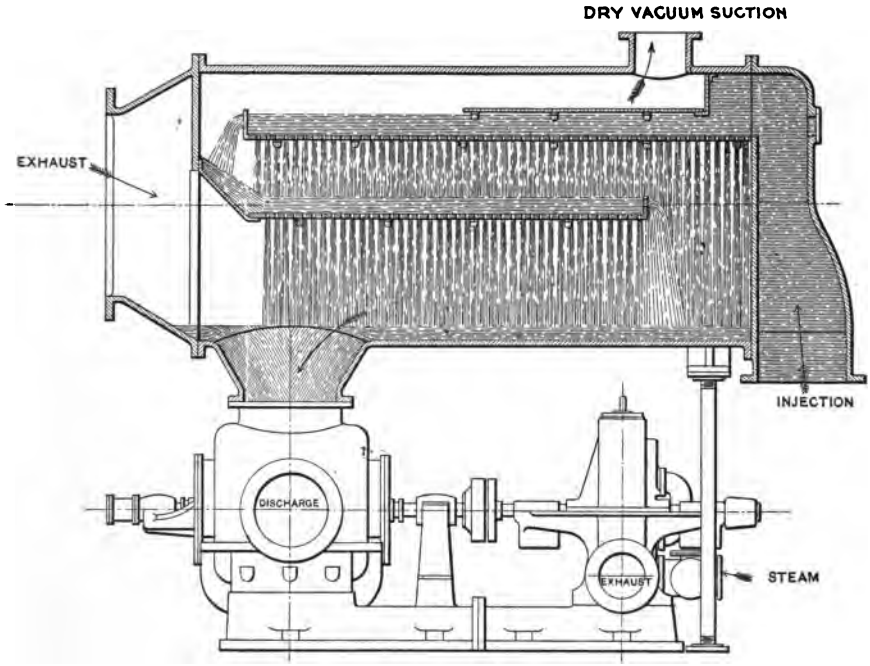

Fig. 134. Worthington Counter-Current Centrifugal Condenser.

Table LXI. gives the approximate floor space required for several sizes of jet condensers of this type based on 70° temperature of condensing water. For other temperatures, larger or smaller sizes must be selected as required.

Worthington Centrifugal Condenser.—Another form of jet condenser, designed to meet the requirements of a high vacuum as demanded by turbine practice, is shown in Fig. 134. The special feature of this condenser is the thorough mixing of the

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steam and water, thus reducing the amount of cooling water required.

The discharge pump used with this condenser is of the centrifugal type driven by a steam turbine.

Condenser Pumps.—Both piston and centrifugal pumps are used for condenser service, the latter being especially well adapted for handling the circulating water.

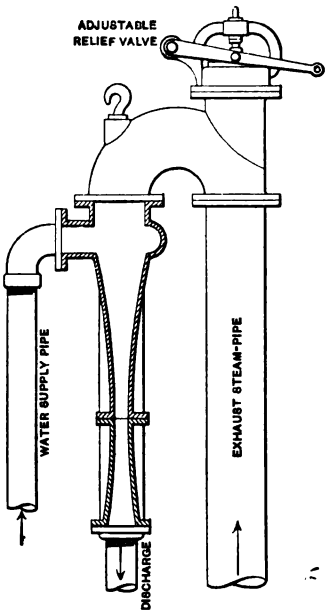

Fig. 135. Knowles Siphon Condenser.

For ordinary vacuums, especially in the case of small and medium sized plants, the air and condensed steam are removed by a single pump as in Figs. 131 and 133.

In the case of turbines, where a high vacuum is required, two pumps are often employed, a "hot-well" pump for removing the condensation, and a "dry-vacuum" pump for the air and vapor. The latter is used in connection with both surface and jet condensers and is connected with the top of the condensing chamber as indicated in Fig. 134.

Knowles Siphon Condenser.—A form of barometric or siphon condenser, is shown in Fig. 135. By a special form of cone the

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injection water is made to take a spiral or rotary motion in its downward flow; this produces a stronger downward suction and is more effectual in drawing the air and vapor through the narrow or middle part of the condenser.

The exhaust steam entering at the top of the condenser passes through the center of the cone and meets the rapidly descending spiral current of water and is immediately condensed.

The water column or discharge pipe between the condenser and the hot-well should be long enough to overcome atmospheric pressure in order to carry the air down into the hot-well after it is drawn through the neck of the condenser.

The supply of injection water can be drawn from a tank or other source after a vacuum is formed, provided the distance above the hot-well is over 10 feet. If the head of water is less than 10 feet, it is necessary to use a pump. A centrifugal or rotary pump is commonly used if the water flows to the pump by gravity, but if it is necessary to raise it from a lower level, a steam or power pump should be used. There are several other forms of condensers, embodying the same principles, but those shown are among the more common, and serve to illustrate the general construction and operation of this piece of apparatus.

Cooling Towers

We have seen from previous computations that from 20 to 30 per cent may be saved in the cost of fuel by the use of a condenser when the injection water can be had free of charge. This, of course, is practically impossible in large towns and cities, unless the plant happens to be located near a river or lake from which the water can be taken. In order to avoid the expense of injection water and still retain the advantages to be derived from a condenser, various methods have been devised, by means of which the discharge water may be cooled and used over and over again. The most common of these is the *cooling tower*, several forms of which are in common use. The general principles involved are practically the same in each, and are very simple.

Principle of the Cooling Tower.—The discharge water as it leaves the condenser is pumped to the top of the tower, where it is distributed upon a series of extended cooling surfaces over

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which it trickles in thin sheets to the bottom and is collected in a reservoir. From here it is taken by a circulating pump and again forced through the condenser. A current of air is made to flow

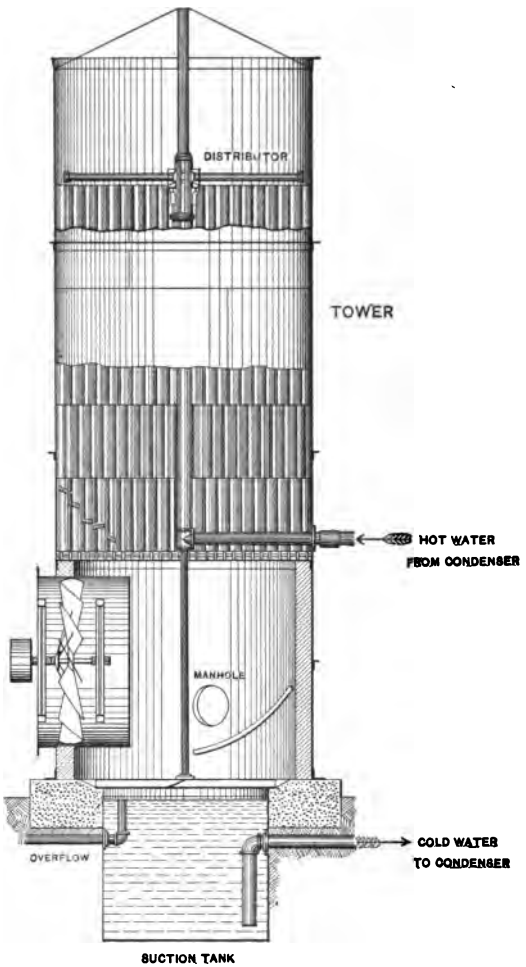

Fig. 136. Worthington Cooling Tower.

over the cooling surfaces, either by the use of a fan or by the chimney effect of the enclosing shell of the tower.

The cooling effect of an apparatus of this kind depends upon three causes. First, radiation from the pipes and sides of the

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tower; second, the contact of cool air with the heated water; and third, evaporation. The last is by far the most important, as the evaporation of a pound of water carries away over a thousand B. T. U.

With the best forms of cooling towers the temperature of the circulating water may easily be reduced 40° to 50° with a loss from evaporation not exceeding 3 or 4 per cent of the total quantity of water passing through the tower.

The Worthington Tower.—The general form of a cooling tower with forced air circulation is well illustrated by the Worthington tower shown in Fig. 136. The cooling surface in this case is made up of pieces of 6-inch terra-cotta pipe in 2-foot lengths placed on end in rows as shown. A steel shell encases the whole, and air is supplied by a disk fan near the bottom. The water to be cooled is pumped to the top of the tower, where it is discharged through a special distributing device and evenly spread over the upper course of the evaporating surface; from here it falls slowly over the successive courses to the cold-water tank at the bottom.

Natural-draft cooling towers are similar in construction except the fan is omitted and the top of the shell is extended in the form of a flue for a considerable distance above the cooling surface.

The tower may be placed just outside the boiler house or upon the roof of the building, as most convenient.

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CHAPTER XIV.

Pipe, Fittings, and Valves

Wrought-Iron Pipe.—Wrought-iron pipe is made "standard weight," "extra strong" and "double extra strong." The standard weight is commonly used for all pressures up to 125 pounds per square inch. The heavier weights are often used for higher pressures, although extensive tests seem to show that standard weight pipe is sufficiently strong for all pressures used in ordinary power work at the present time.

When particularly exposed to corrosion, it is well to use extra heavy pipe. This applies to feed lines, sealed returns, underground drip piping, etc., and all lines that are to be run in places which are not easily accessible.

Nearly all of what is commonly known as wrought-iron pipe is in reality wrought steel. There seems to be no especial advantage in using iron in place of a good quality of steel, although some engineers prefer the former.

Wrought-iron pipe is not carried in stock to any great extent, and if it is to be used in any considerable quantity should be ordered well in advance of the time it is to be used.

Table LXII. gives the dimensions of standard wrought-iron pipe.

The outside diameter of the extra strong and double extra strong pipe is the same as that of the standard weight, the extra thickness being added to the inside. This makes it possible to use the same fittings with any weight of pipe.

Brass and Copper Pipe.—Seamless drawn brass and copper tubing may be had in sizes corresponding to standard wrought iron pipe.

This material is used for hot-water service, for connections between feed pumps and boilers, for the attachment of boiler trimmings, and for steam-heating coils in hot-water boilers. It is more durable than iron pipe when in contact with hot water, and its freedom from rust makes it especially adapted for hot-water service in connection with laundry and similar work.

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Tin Lined Pipe.—For long lines of water piping where the expense of brass piping would be too great, iron pipe lined with an alloy of tin and lead is often used. This kind of piping is especially adapted for use in chemical works, where a lead lining is used in place of the alloy.

TABLE LXII.*

DIMENSIONS OF STANDARD WROUGHT-IRON PIPE.

¼ and smaller proved to 800 pounds per square inch by hydraulic pressure.
1¼ and larger proved to 500 pounds per square inch by hydraulic pressure.

Nominal diameter	Actual outside diameter	Actual inside diameter	Inside circumference	Outside circumference	Length of pipe per square foot of inside surface	Length of pipe per square foot of outside surface	Inside area	Outside area	Length of pipe containing one cubic foot	Weight per foot
In.	In.	In.	In.	In.	Ft.	Ft.	In.	In.	Ft.	Lb.
⅛	0.405	0.270	0.848	1.272	14.15	9.44	0.0572	0.129	2500.	0.243
¼	0.54	0.364	1.144	1.696	10.50	7.075	0.1041	0.229	1385.	0.422
⅜	0.675	0.494	1.552	2.121	7.67	5.657	0.1916	0.358	751.5	0.561
½	0.84	0.623	1.957	2.652	6.13	4.502	0.3048	0.554	472.4	0.845
¾	1.05	0.824	2.589	3.299	4.635	3.637	0.5333	0.866	270.	1.126
1	1.315	1.048	3.292	4.134	3.679	2.903	0.8627	1.357	166.9	1.670
1¼	1.66	1.380	4.335	5.215	2.768	2.301	1.496	2.164	96.25	2.258
1½	1.90	1.611	5.061	5.969	2.371	2.01	2.038	2.835	70.65	2.694
2	2.375	2.067	6.494	7.461	1.848	1.611	3.355	4.430	42.36	3.600
2½	2.875	2.468	7.754	9.032	1.547	1.328	4.788	6.491	30.11	5.773
3	3.50	3.067	9.636	10.996	1.245	1.091	7.388	9.621	19.49	7.547
3½	4.00	3.548	11.146	12.566	1.077	0.955	9.887	12.566	14.56	9.055
4	4.50	4.026	12.648	14.137	0.949	0.849	12.730	15.904	11.31	10.66
4½	5.00	4.508	14.153	15.708	0.848	0.765	15.939	19.635	9.03	12.34
5	5.563	5.045	15.849	17.475	0.757	0.629	19.990	24.299	7.20	14.50
6	6.625	6.065	19.054	20.813	0.63	0.577	28.889	34.471	4.98	18.767
7	7.625	7.023	22.063	23.954	0.544	0.595	38.737	45.663	3.72	23.27
8	8.625	7.982	25.076	27.096	0.478	0.444	50.039	58.426	2.88	28.177
9	9.625	9.001	28.277	30.433	0.425	0.394	63.633	73.715	2.26	33.70
10	10.75	10.019	31.475	33.772	0.381	0.355	78.838	90.762	1.80	40.06
11	12.00	11.25	35.343	37.699	0.340	0.318	98.942	113.097	1.455	45.95
12	12.75	12.000	38.264	40.840	0.313	0.293	116.535	132.732	1.235	48.98
14	14.00	13.25	41.268	43.982	0.290	0.273	134.582	153.938	1.069	53.52
15	15.00	14.25	44.771	47.124	0.271	0.254	155.968	176.715	.923	57.89
16	16.00	15.25	47.874	50.265	0.254	0.238	177.867	201.062	.809	61.77
18	18.00	17.25	53.281	56.548	0.225	0.212	225.907	254.469	.638	69.66
20	20.00	19.25	59.288	62.832	0.202	0.191	279.720	314.160	.515	77.57

* From catalogue of Walworth Manufacturing Company.

Spiral Riveted Pipe.—Spiral riveted galvanized iron pipe is quite frequently used for exhaust lines and low-pressure heating mains of large size. It has the advantage of lightness, and is lower in cost than standard wrought-iron pipe. It is made up with flanged fittings for steam work the same as other kinds of

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pipe. Fig. 137 shows a section of this pipe and the method of making up the joints for steam and hot-water work. It is commonly made in lengths of 15, 20 and 25 feet.

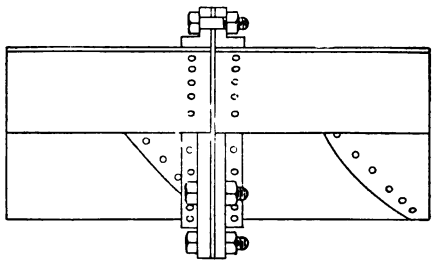

Fig. 137. Spiral Riveted Pipe.

Equation of Pipes.—It is frequently desired to know what number of pipes of a given size are equal in carrying capacity to one pipe of a larger size. At the same velocity of flow the volume

TABLE LXIII.

EQUATION OF PIPES.

Diameter, inches	1	2	3	4	5	6	7	8	9	10	12	14	16	18	20
2	5.7	1													
3	15.6	2.8	1												
4	32	5.7	2.1	1											
5	55.9	9.9	3.6	1.7	1										
6	88.2	15.6	5.7	2.8	1.6	1									
7	130	22.9	8.3	4.1	2.3	1.5	1								
8	181	32	11.7	5.7	3.2	2.1	1.4	1							
9	243	43	15.6	7.6	4.3	2.8	1.9	1.3	1						
10	316	55.9	20.3	9.9	5.7	3.6	2.4	1.7	1.3	1					
11	401	70.9	25.7	12.5	7.2	4.6	3.1	2.2	1.7	1.3					
12	499	88.2	32	15.6	8.9	5.7	3.8	2.8	2.1	1.6	1				
13	609	108	39.1	19	10.9	7.1	4.7	3.4	2.5	1.9	1.2				
14	733	130	47	22.9	13.1	8.3	5.7	4.1	3.0	2.3	1.5	1			
15	787	154	55.9	27.2	15.6	9.9	6.7	4.8	3.6	2.8	1.7	1.2			
16	.....	181	65.7	32	18.3	11.7	7.9	5.7	4.2	3.2	2.1	1.4	1		
17	.....	211	76.4	37.2	21.3	13.5	9.2	6.6	4.9	3.8	2.4	1.6	1.2		
18	.....	243	88.2	43	24.6	15.6	10.6	7.6	5.7	4.3	2.8	1.9	1.3	1	
19	.....	278	101	49.1	28.1	17.8	12.1	8.7	6.5	5	3.2	2.1	1.5	1.1	
20	.....	316	115	55.9	32	20.3	13.8	9.9	7.4	5.7	3.6	2.4	1.7	1.3	1

* From the catalogue of the Walworth Manufacturing Company.

delivered by two pipes of different sizes is proportional to the squares of their diameters; thus, one 4-inch pipe will deliver the same volume as four 2-inch pipes. With the same head, however, the velocity is less in the smaller pipe, and the volume delivered varies about as the square root of the fifth power. Table LXIII.

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has been calculated on this basis. The figures opposite the intersection of any two sizes is the number of the smaller-sized pipes required to equal one of the larger. Thus, one 4-inch pipe is equal to 5.7 2-inch pipes.

Pipe Fittings

Cast-iron fittings, both screwed and flanged, are used for all classes of steam work. These are commonly made in three weights. The lightest for low pressures, such as exhaust or condenser connections; standard weight, for pressures up to 100 or 125 pounds per square inch; and extra heavy for higher pressures up to 250 pounds per square inch.

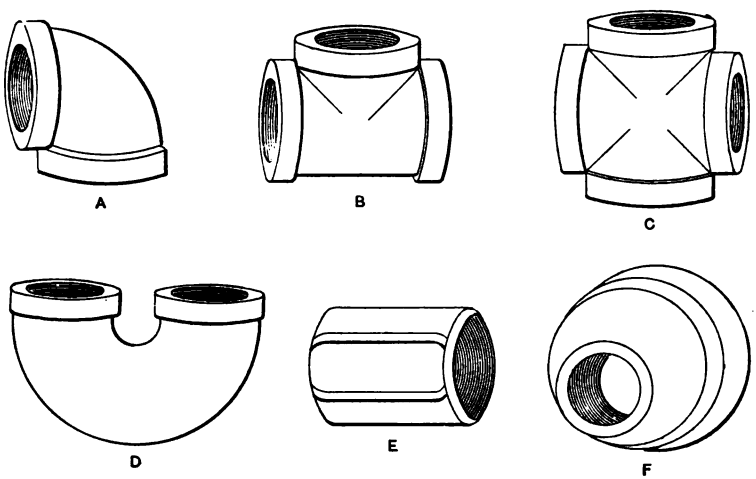

Fig. 138. Pipe Fittings.

Fittings are made in a great variety of forms, the most common being the elbow, tee, cross, return bend and coupling, shown in Fig. 138, *A* to *E* inclusive. At *F* is an eccentric reducing coupling for use when connections are made in the side or top of a long run of pipe, which reduces in size as the branches are taken off. By the use of this fitting the bottom of the main is kept level and pockets for the collection of water are avoided.

Various modifications of these fittings may be had to meet almost any condition which may arise in designing a system of piping.

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Elbows of 45 degrees are used for making an eighth turn, and Y branches for making a 45-degree connection with a main.

When pipes are to be joined permanently, couplings are commonly used, but if it is desired to make any part of the work removable, unions are employed instead. All steam and water connections with boilers, engines, pumps and other apparatus should be made up with unions. For small pipes, the screwed brass union shown in Fig. 139 is used, but for larger sizes, over 2 inches,

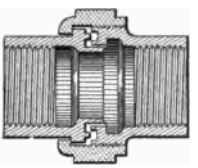

Fig. 139. Union.

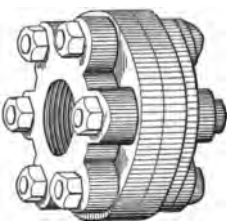

Fig. 140. Flange Joint.

cast-iron flanges are screwed on to the ends of the pipe and then bolted together as indicated in Fig. 140.

Pipe Bends.—Pipe bends of various forms are commonly used in making connections about boilers and engines for taking the strains due to expansion. Connections made up in this way are less rigid than when straight lengths of pipe and cast-iron fittings are used. Pipe bends are often employed in hot-water or steam work where it is desired to reduce the friction in the pipe to the smallest amount.

Fig. 141 shows some of the more common forms. These bends are made to order with a minimum radius equal to 5 diameters of the pipe. From 4 to 6 inches of straight pipe should be allowed at each end next to the flange.

Joints, Hangers, Etc.

Flanged Joints.—For low steam pressures, the flanges are simply screwed onto the pipe ends, a thin gasket of sheet packing placed between them, and then drawn together with bolts. For higher pressures more care must be taken to make the joints tight and durable. Flanges are commonly made in three ways, as follows: Those with a plain face, Fig. 142; tongue and groove,

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Fig. 143, and with a raised face inside the bolt holes, Fig. 144. There are various methods of securing the flange to the pipe in order to make a tight joint. One of the simplest and most satisfactory, considering first cost, ease of repairing, and freedom

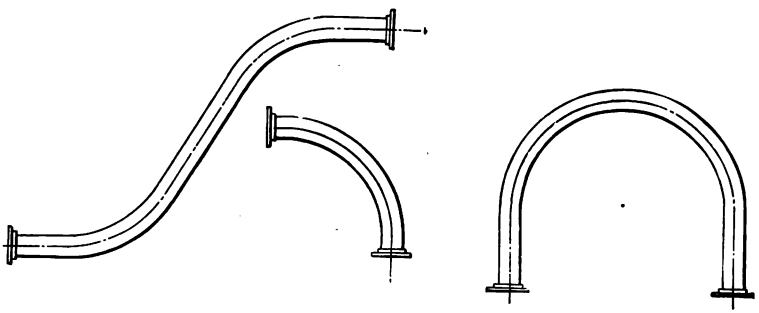

Fig. 141. Pipe Bends.

from leaks, is made by threading the pipe with a full taper, then screwing on the flange by power until the end of the pipe projects about 1/16 inch, and facing off in a lathe. Another method is to round off the inner edge of the flange and expand or peen over

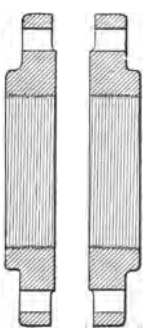

Fig. 142.

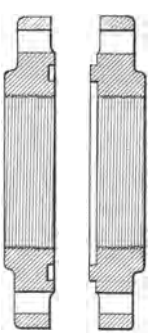

Fig. 143.

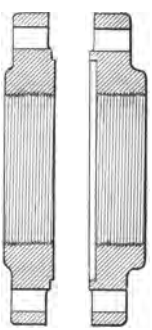

Fig. 144.

the end of the pipe as shown in Fig. 145 and then face in a lathe as before. Flanges are sometimes shrunk in place, and also riveted. With the latter arrangement the joint is apt to leak after a time around the rivets. Where especially fine work is desired, and the first cost does not prohibit it, wrought-iron pipe may be

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used with flanges of the same material welded in place. Work of this kind must be done where the pipe is manufactured.

There are several patented joints which give very satisfactory results. That shown in Fig. 146 is made by the Walworth Manufacturing Company. Its construction is evident from the cut and requires no explanation. The principal objection to the tongue

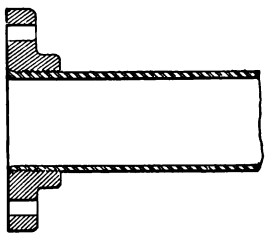

Fig. 145.

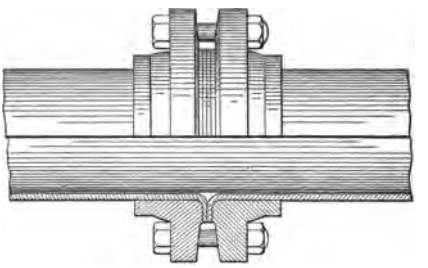

Fig. 146.

and groove joint, and also to the raised face, when the depth of step is too great, is the difficulty of springing apart a line of heavy piping to remove a section, or for repacking a joint.

All flanged joints, except in special cases, should be provided with gaskets or packing of a material and construction suitable for the pressure to be carried in the piping.

In the case of low pressure work these are usually cut from sheet packing and made to cover the entire face of the flange.

High-pressure packing is commonly made up in the form of rings or gaskets which fit inside the bolt circle. When flanges of the tongue and groove type are used, special gaskets are required.

Sheet packing is made in various grades, the best being composed of rubber and other heat resisting materials properly mixed and vulcanized. Some makes also contain a metallic fabric embedded in the sheet.

For high-pressure work, with faced flanges, the packing should not be over 1/16 inch in thickness; for low-pressure work, and where the flanges are more or less uneven, it should be about twice that thickness. In some of the highest class of work, ground joints are employed without the use of packing.

Hangers.—Hangers of various patterns are used for supporting

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steam and return pipes in overhead positions. When the ceiling construction is of wood they are usually held in place by lag screws, screwed into the beams or joists. When fireproof construction is used, the hangers are either clamped to the lower

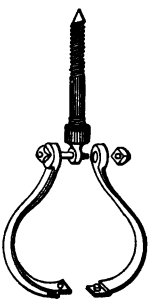

Fig. 147.

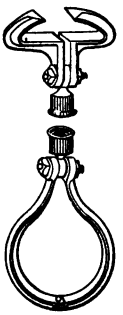

Fig. 148.

Pipe Hangers.

flanges of the *I*-beams or attached to iron plates embedded in the masonry above the arches. The requirements of a satisfactory hanger are that it shall be adjustable to the proper alignment of the pipe and have sufficient lateral movement to swing with the

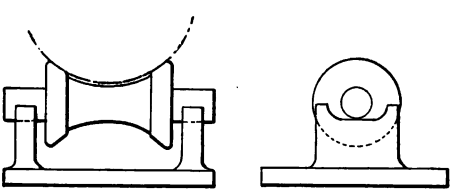

Fig. 149. Pipe Support.

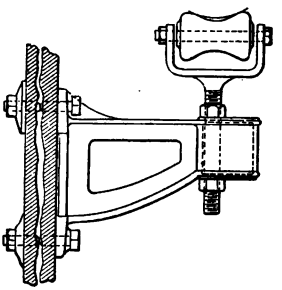

Fig. 150. Pipe Bracket.

expansion of the pipe without straining. Hangers are usually placed from 10 to 12 feet apart on straight runs of pipe, with extra ones at bends or where other special conditions require them. Fig. 147 shows an adjustable hanger with lag screw for attaching to a wooden ceiling, while Fig. 148 is provided with a clamp for fastening to an *I*-beam. Where pipes run near the floor they are usually supported on rolls which rest in small cast-iron bases screwed or bolted to the floor. (See Fig. 149.)

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Fig. 150 shows a similar roll attached to a wall bracket. Where several overhead pipes are run side by side they are often supported upon rolls turning upon rods hung from the ceiling above.

Spring hangers are sometimes employed when piping connecting with high-speed engines is supported from the floor above, in buildings where vibration is objectionable. In case hangers of this kind are used, some form of safety device should be employed for catching the pipe should the springs give way. The constant vibration of the piping is almost sure to weaken the springs after a time and other means of support for use in case of an emergency should always be provided.

Pipe Expansion

The following table shows the expansion of wrought-iron pipes when steam at different pressures is turned into them.

TABLE LXIV

EXPANSION OF WROUGHT-IRON PIPE DUE TO HEAT FROM STEAM.

Temperature of air when pipes are fitted	Expansion per 100 feet wnen heated to			
	215 degrees, 1 lb. steam pressure	265 degrees, 25 lb. steam pressure	297 degrees, 50 lb. steam pressure	338 degrees, 100 lb. steam pressure
0 degrees	1.72 inches	2.12 inches	2.31 inches	2.70 inches
32 "	1.47 "	1.78 "	2.12 "	2.45 "
64 "	1.21 "	1.61 "	1.87 "	2.19 "

From the above it is evident that in laying out a system of piping care must be taken to provide for this increase in length without producing excessive strains upon the pipe and fittings. In case of the smaller sizes this may be cared for by means of offsets and bends which allow the pipe to spring or give. For the larger sizes, this simple method will not be sufficient unless the offsets are of considerable length, and swivels or slip joints must be used to take up the expansion.

Expansion Joints.—The method of making up a swivel joint is shown in Fig. 151. Any lengthening of the pipe *A* will be taken up by a slight turning or swivel movement at the points *B* and *C*. The length of the offset *B C* should be at least 5 feet

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for each 100 feet length of run. Special joints are also made which may be used in place of the elbows and nipples shown in Fig. 151.

If slip joints are used on high-pressure work they should

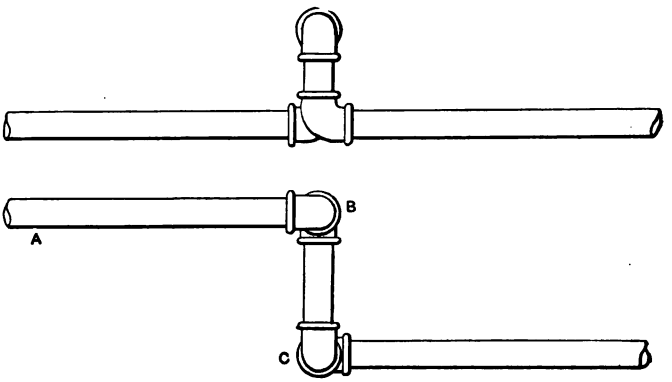

Fig. 151. Swivel Joint.

always be balanced against the steam pressure acting upon the end of the pipe. Fig. 152 illustrates an effective form of balanced ex-

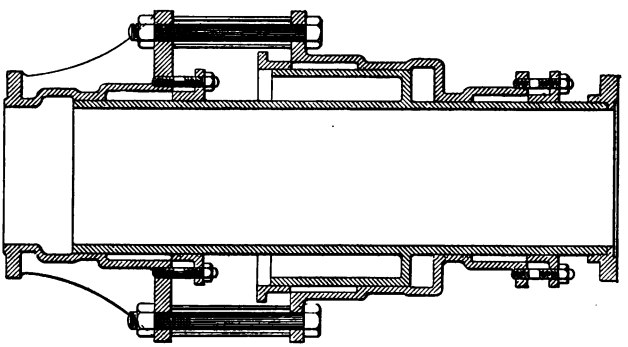

Fig. 152. Balanced Expansion Joint.

pansion joint. It has an enlargement cast on the sleeve, which forms a piston, as shown. In the sleeve below the piston are drilled holes which admit live steam. The area of the piston on which the steam acts must be made equal to the area of the main steam pipe, so that, while permitting free expansion, it is also perfectly balanced.

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The expansion loop shown in Fig. 153 is used to some extent for pipes up to 3 or 4 inches in diameter where the amount of expansion is small. Copper was formerly used to quite an extent for loops of this kind, but on account of its low elastic limit, and the liability of flaws in manufacture it has been largely superseded by the best quality of mild steel.

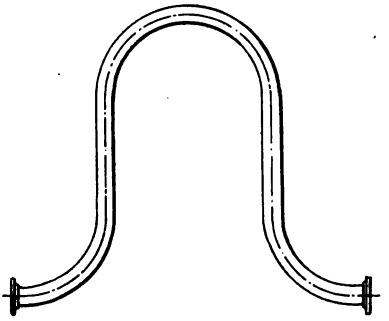

Fig. 153. Expansion Loop.

Valves

Among the requirements of a good valve, are, sufficient weight of metal to prevent its being bent or sprung out of shape when connected with the piping; valve seats that are easily repaired or renewed, freedom from pockets or projections which may catch dirt or scale, and arrangements for easily packing the stem or spindle when under pressure.

Gate Valves.—This type of valve gives very little resistance to the flow of steam or liquid passing through it, and is generally used in the best class of work. The general construction of the Chapman valve is shown in Figs. 154 and 155, which represent the "inside screw" and "outside screw" or yoke patterns, respectively. In the former the spindle remains stationary so far as any vertical movement is concerned, and the gate or plug being attached to it by means of a threaded nut, rises into the bonnet when the spindle is revolved. With this form of valve it is impossible to tell by its appearance whether it is open or closed unless provided with some special device for indicating the open and closed positions. With the outside screw, the upper portion of the spindle is threaded and is operated by a revolving nut *N*, held

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vertically in the yoke *Y*, and turned by the hand wheel which is fastened to it. The operating screw of these valves is entirely outside the valve body where it can be inspected and oiled, thus insuring easy operation. The spindle rises through the stuffing box without turning, which allows the packing to be kept tight without difficulty. When valves of this form are used, the attendant can tell at a glance whether they are open or closed.

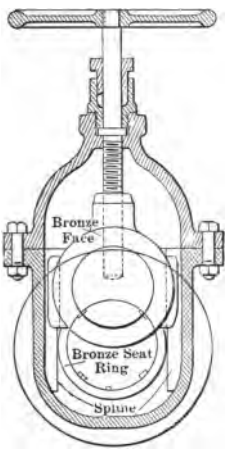

Fig. 154. Inside Screw.

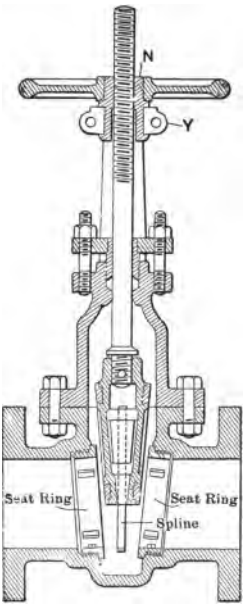

Fig. 155. Outside Screw.

Gate Valves.

The gate is made wedge shaped and presses against ground seats when closed. The seats are made both solid and removable, and of various materials for different pressures. Gate valves of different makes vary chiefly in the small details of construction.

In the Ludlow valve the plug or gate is made up of two parts having the outer faces parallel instead of being wedge shaped as in the case of the Chapman valve. The backs of these two parts are inclined, so that when the valve is shut, the plug first descends and loosely closes the opening; then a further movement of the

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stem forces the wedges slightly past each other and presses the outer surfaces tightly against the seats.

In certain forms of the Fairbanks valve the distinctive feature is a metal cage which holds the seat rings in place. With valves of this form, the rings can be renewed by simply taking off the bonnet and without removing the valve from the pipe line. A special seat ring of vulcanized asbestos is used in this type of valve.

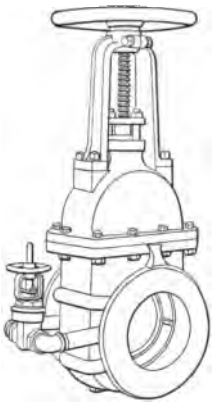

Fig. 156. By-pass Valve.

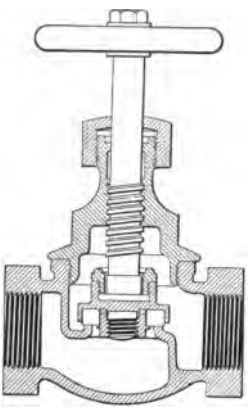

Fig. 157. Globe Valve.

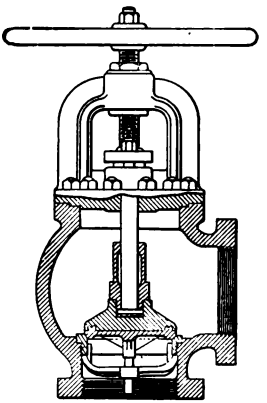

Fig. 158. Angle Valve

Choosing a Valve.—Care should always be taken to select a valve in which the material of the seat or ring is suited to the pressure to be carried. Bronze seats are commonly used for pressure over 80 pounds, unless the babbitt metal or other material employed is especially prepared for higher pressures and guaranteed by the makers.

For reasons already stated, outside screw valves are preferable for high-pressure work, and especially so for the larger sizes. In the case of low-pressure heating work, the inside screw is more commonly used.

Locating Gate Valves.—A gate valve should never be placed in a steam pipe with the spindle downward, for if only partially opened, the gate projecting from the bottom upward, will form a pocket for holding back the condensation, and may thus be the cause of serious injury to the plant. When placed with the

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stem extending upward or in a horizontal position a clear passage is provided at the bottom of the pipe. Valves are made in different weights and the pressure to be carried should always be stated when they are specified.

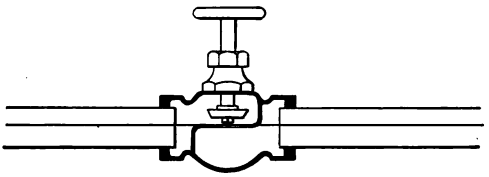

Fig. 159.

By-pass Valves should be used in high-pressure work for sizes 8 inches in diameter and over. A by-pass valve is shown in Fig. 156. There are two advantages in the use of by-pass valves in

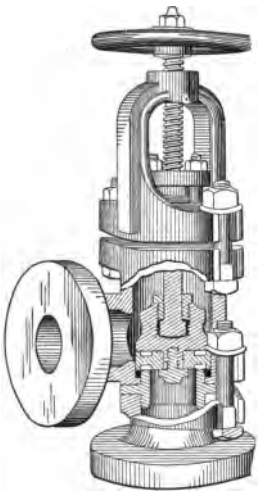

Fig. 160. Cadman Blow-off Valve.

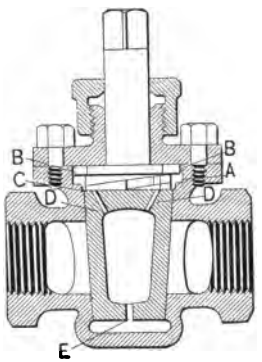

Fig. 161. Homestead Blow-off Cock.

high-pressure lines. First, when the valve is closed, there is an enormous pressure acting on one side of the gate which makes it very hard to start, and also as it is pressed so tightly against the seat on the opposite side, it is apt to score the smooth surfaces of the gate and seat when opened. When used in the supply pipe to an engine it is desirable to admit steam very slowly when

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"warming up," and this is much more easily accomplished by means of a small by-pass valve which is easily operated, than by the use of the main valve itself.

Globe Valves.—A form of globe valve is shown in Fig. 157. This is known as a disc valve, and has a renewable seat; other patterns have solid seats of different forms. A globe valve should always be set to close against the pressure, for if placed in the opposite way it could not be opened if the valve should become detached from the stem, and no indication of the trouble would be given by the action of the stem under these conditions.

Globe valves should never be placed in a horizontal steam or dry return pipe with the stem in a vertical position. The reason for this is made evident in Fig. 159 which shows by the middle line the height to which the condensation must rise before it can flow through the port or opening.

Angle Valves.—These valves are sometimes used to take the place of elbow fittings. They are a form of globe valve with the inlet at the bottom and the outlet in the side. The larger sizes have a guide spindle extending below the valve seat as shown in Fig. 158, while in the smaller sizes this is omitted.

Blow-off Valves and Cocks.—The blow-off pipe of a boiler is usually provided with a plug cock, having a gate valve placed between it and the boiler as a safeguard in case of leakage. Special blow-off valves are sometimes used in place of the cock and valve. These are made very heavy and are so designed that it is difficult for dirt or scale to lodge beneath the seat. Fig. 160 shows the Cadman blow-off valve which is constructed upon these principles.

The difficulty experienced with the ordinary plug or blow-off cock is its liability to leak unless set down very tightly against its seat, in which case it is apt to stick, especially when first starting it. To overcome this difficulty, special forms have been designed, among which the Homestead locking cock shown in Fig. 161 is a good illustration. This is provided with a cam *A*, so arranged that after the cock is closed a further movement of the stem in the same direction forces the plug firmly against its seat. A slight movement in the opposite direction releases the cam and allows the plug to turn easily to the open position. The balancing ports *E, D, D* allow the pressure to enter the spaces

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above and below the plug, and as the upper end is the larger the pressure upon it is slightly greater, and therefore holds the plug lightly against its seat while open.

Check Valves.—When it is necessary that the flow of steam or water shall always take place in the same direction, check valves are employed. There are several forms of these in use, the most common of which is the swing check shown in Fig. 162.

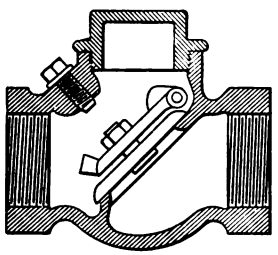

Fig. 162. Check Valve.

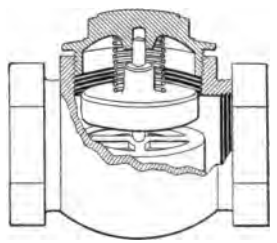

Fig. 163. Spring Check.

The seat in this valve is placed at an angle of about 45 degrees to the direction of flow. The valve or "clapper" is fitted somewhat loosely where it is fastened to the swinging arm, so that it may properly seat itself. This form is usually preferred in heating work as it gives less resistance to the flow of steam or water, and by its construction offers but little opportunity for scale or sediment to lodge upon the seat. When it is desired to reduce the resistance to a minimum, the check may be turned partially on its side or a special aluminum valve may be used.

The Locke spring check shown in Fig. 163 is especially adapted for use in boiler feed pipes where the feed-water is supplied by a pump. The valve being held in place by a light spring is prevented from beating against the seat between the strokes of the pump. Check valves are placed in boiler-feed and return pipes, in the discharge pipes from traps, and various other places where it is desired to prevent the pressure in one pipe from "backing" into another, as when several return pipes from a building are brought into a common receiving tank. When used in a return pipe the check should always be placed below the water line if possible.

Pressure Reducing Valves.—It is often necessary to provide steam at different pressures in the same building, as in the case of a combined power and heating plant. In this case the re-

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duction in pressure is accomplished by passing the steam through a reducing valve. There are many different forms of these valves, all acting more or less upon the same general principles. In the Kieley valve, shown in Fig. 164, the low pressure steam acts on the lower side of a flexible diaphragm, and the weighted lever, which may be adjusted to give the desired reduction in pressure, acts upon the other side. The movement of this diaphragm causes

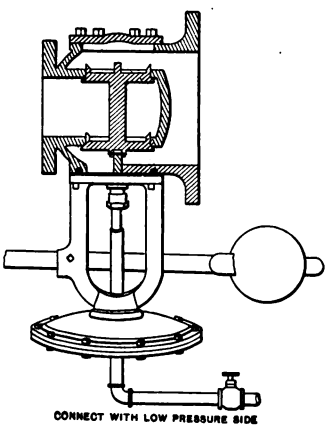

Fig. 164. Kieley Reducing Valve.

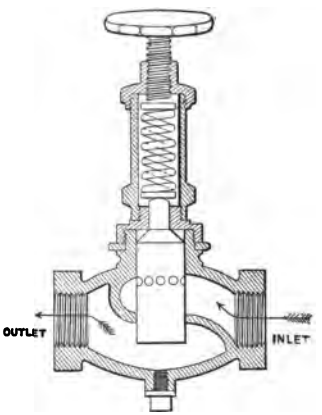

Fig. 165. Watson Reducing Valve.

a balanced valve to open or close as may be necessary to maintain the desired lower pressure.

Another form, in which the weighted lever is replaced by a spring and the diaphragm by a piston valve, is illustrated by the Watson reducing valve shown in Fig. 165. In this case the reduced pressure acts upon the lower end of a piston valve which is balanced between this pressure and an adjustable spring at the top. As the piston simply passes through the inlet chamber, its action is not affected by the high-pressure steam surrounding it. There are several good valves in use which embody the same principles or a combination of the two. Those shown are among the simplest and are therefore well adapted for purposes of illustration.

When the reduction in pressure is large, as in the case of a combined power and heating plant, the valve may be one or two sizes smaller than the low-pressure main into which it discharges.

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For example—a 5-inch valve will supply an 8-inch main, a 4-inch a 6-inch main, a 3-inch a 5-inch main, a 2½-inch a 4-inch main, etc.

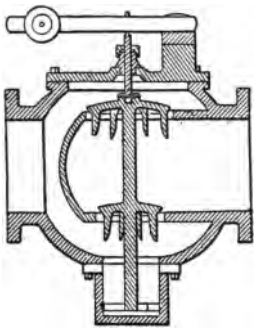

Fig. 166. Kieley Back-Pressure Valve.

For the smaller sizes the difference should not be more than one size. All reducing valves should be provided with a valved by-pass and cut-out valves for use in case of repairs.

Back Pressure Valve.—This is a form of relief valve which is

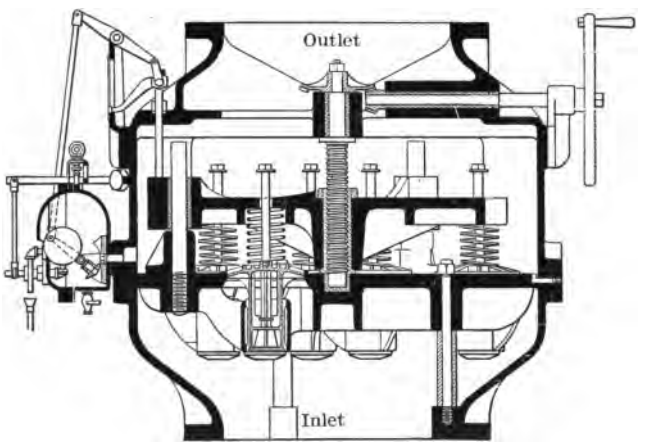

Fig. 167. Cross Section of Cochrane Exhaust Outlet Valve.

placed in the outboard exhaust pipe from an engine to prevent the pressure in the heating system from rising above a given

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point. Its office is the reverse of the reducing valve which supplies more steam when the pressure becomes too low.

The Kieley valve is illustrated in Fig. 166. The essential features of this valve are two disks, of different size, attached to a common spindle. The force tending to open the valve is the steam pressure acting upon the difference in area of the disks.

Adjustment is secured by varying the position of the counterweight upon the lever arm.

Another type of relief valve, especially adapted to pipes of large diameter, is shown in Fig. 167. This contains a valve deck upon which are placed a number of small spring valves, each provided with a dash-pot for cushioning its motion. The tension of the springs, and consequently the pressure at which the valves open, is determined by the position of an adjustable pressure plate operated from the outside. A back-pressure valve is simply a low-pressure safety-valve, provided with an especially large opening for the passage of steam through it.

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CHAPTER XV.

DETAILS OF POWER PLANT DESIGN

Boilers and their equipment have already been taken up in some detail and will therefore not be included in the present chapter. This will be devoted largely to the selection of steam engines for different purposes, the laying out of steam and feed piping, and other details of design.

The Selection of a Steam Engine

The selection of an engine for any particular location depends upon the conditions under which it is to operate. For sizes under 100 horse power, and especially where the exhaust steam is to be used for heating purposes, the simple non-condensing engine is generally used. For larger sizes, and where the steam pressure is over 120 pounds, the compound engine is often advocated upon the ground of economy.

The first cost of a compound engine is considerably more, being perhaps 30 per cent greater for the average machine. The steam consumption in the case of high-speed non-condensing engines at full load, is from 20 to 30 per cent less for the compound, while the increase in fuel consumption required to raise the boiler pressure from 80 to 125 pounds is only about 1 per cent. When all of the exhaust is used for heating, there is of course nothing to be gained by installing a high-priced engine for the sake of reducing the steam consumption. But on the other hand if the exhaust can only be used for a comparatively small part of the year, or if but a portion of it can be utilized in the heating system, then the saving will often offset the interest and depreciation changes upon the additional first cost and leave a good margin of profit besides, this being especially true in localities where the cost of fuel is high.

Use of Compound Engines.—Compound engines have been growing in favor of late years for use in connection with heating systems, and in many cases have shown their superiority when "all-the-year-round" economy is considered. There may be a

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period during the coldest months when no gain is shown over the simple engine, but when the milder weather of spring and fall is considered, together with the summer months, when no use can be made of the exhaust steam except for heating the feed-water, the economy of the compound engine becomes evident. The main point is to always operate the compound unit within its economical range, that is, from 50 per cent load to full load, as at light loads the gain falls off quite rapidly.

The best type of engine as regards speed, depends much upon the available room. In office buildings and other similar locations where floor space is valuable, the high-speed engine is widely used.

For central lighting plants in connection with public institutions, or where floor space is not so limited, the moderate-speed engine is a good type to employ.

For factory and general power work, the slow-speed engine of Corliss or similar design is generally used, being economical in the use of steam, durable, and easily cared for.

Engines for Operating Dynamos.—As the engines in small and medium sized plants are used largely for driving electric generators, it is necessary to understand the method of changing mechanical energy into electrical, and vice versa. In order to make these computations more clearly understood, the following definitions are given:

Electro-motive Force.—When a difference of electrical potential, or pressure, exists between two points, there is said to exist an *electro-motive force,* or a tendency for a current to flow from one point to the other. In a voltaic cell, one plate is at a different potential from the other, which gives rise to an electro-motive force between them. In a dynamo a difference of potential exists between the two poles, called *voltage.*

Current.—A current of electricity flows when two points at a different potential are connected by a wire or other conductor.

Resistance.—That property of matter which opposes or resists the flow of electricity through it is called *resistance.*

The units of electro-motive force, current, and resistance, are the *volt, ampere,* and *ohm,* respectively, and are defined in terms of one another as follows:—

An ampere is the amount of current which will flow through

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a resistance of one ohm when the electro-motive force is one volt.

An ohm is the resistance of a conductor through which a current of one ampere will pass when the electro-motive force is one volt.

A volt is the electro-motive force required to cause a current of one ampere to flow through a resistance of one ohm.

Power.—Power is the rate of doing work and expresses the amount done in a given time.

Joule.—A *joule* is the electrical unit of work, and is equivalent to the energy expended in one second by a current of one ampere passing through a resistance of one ohm.

Watt.—A *watt* is the unit of power, and is equivalent to the work done at the rate of one joule per second.

The *Kilowatt,* which is equal to 1,000 watts, is the unit employed almost exclusively in practical work. Its symbol is Kw.

The *Kilowatt-hour* is a common unit of energy, and represents the work done in one hour when the power acting is one Kilowatt.

Equivalence of Electrical Energy in Mechanical Units.—The common unit of mechanical energy is the foot pound, and from experiment it has been found that 1 joule is equivalent to 0.7373 of a foot pound; that is, the same amount of heat will be developed by 1 joule as by 0.7373 of a foot pound of work. As 1 horse power is equal to 550 foot pounds of work per second, it follows that it is also equal to $\frac{550}{0.7373} = 746$ joules per second.

According to the definitions previously given, energy expended, or work done, at the rate of 1 joule per second is equivalent to 1 watt of power; hence, if 746 joules per second equal 1 horse power, then its equivalent, 746 watts, must equal 1 horse power.

Therefore, to find the equivalent of mechanical power in electrical power, multiply the horse power by 746, and to find the equivalent of electrical power in mechanical power, divide the number of watts by 746, or more briefly:

$$\text{horse power} \times 746 = \text{watts},$$
$$\text{watts} \div 746 = \text{horse power}.$$

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Practical Considerations

Losses in Energy.—In operating an electrical machine there is a certain loss in energy due partly to the friction of the mechanical parts, and partly to the resistance to the current. The ratio of the amount of energy which a machine gives out to that which it receives is called the *commercial efficiency*. For example, if an input of 100 horse power of mechanical energy is given to a generator and only 90 horse power of electrical energy is delivered, the efficiency is $(90 \times 100) \div 100 = 90$ per cent. The efficiency of generators varies according to their size and the load carried.

Tests of a standard make show an efficiency of from 85 to 95 per cent for generators ranging from 5 to 200 Kw. at full load, and an efficiency of 82 to 87 per cent for the same machines at one half load.

Number of Units.—A dynamo with its connecting engine is called a *unit*. In an electric lighting plant the load is usually very light during the day and is at a maximum for a few hours only during the evening. For this reason it is very desirable, unless the power supplied is small, to have several units so arranged that as the load increases their power may be successively added as required. The relative size of the units is a matter upon which there is some difference in opinion. The best economy in operation is obtained where large machines are used during periods of heavy loads and those of small size during light loads.

While greater economy may be secured by making all the units of a different size, certain practical considerations may often lead to a different course.

It is the custom of many engineers to divide the total capacity into three units, each of such size that two of them, by being overworked not exceeding 30 per cent, can do the maximum work satisfactorily. This permits the withdrawal of one unit in case of accident or repairs, and besides, the three are interchangeable on the work. Their first cost is usually less, they adapt themselves better to the available space in the engine room, only one size of parts need be kept for repairs, and in case of accident the plant would not be crippled as would be the case if the units were of different size and the largest should break down at the time of heaviest loading. The size of the dynamo in connection

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with light loads, is not of so much importance as that of the engine, for its efficiency does not decrease so rapidly with underloading. Even at one third load the efficiency of some dynamos is but little less than that at full load; also small dynamos are made nearly as efficient as those of larger capacity, so that it is allowable to suit their size and number to the requirements of the engines.

Direct Coupling of Engines and Dynamos.—This has many advantages over other methods of connecting. It is simple, compact, produces no loss of efficiency due to belting or countershafting, and does away with the noise of belt-driven machines. The economy in space is also of much importance in the case of office and other city buildings.

While the belted type is still used occasionally, under special conditions, the direct-connected unit is almost universally employed in the equipment of new plants.

Power of Engine.—The size of engine required for driving a dynamo depends upon the efficiencies of both machines. For full loads, under ordinary conditions, we may assume an efficiency of 80 to 85 per cent for engines up to 100 horse power, and 85 to 90 per cent for larger sizes; and efficiencies of 85 to 90 per cent for generators up to 100 Kw. and 90 to 95 per cent for larger machines.

Example.—What must be the indicated horse power of an engine to drive a dynamo delivering 100 Kw.?

As the I. H. P. of the engine must always be greater than the Kw. rating of the dynamo, it is evident in this case that it must be over 100, and we can therefore assume an efficiency of 85 per cent. We have 100 Kw. = 100,000 watts = 100,000 ÷ 746 = 134 H. P. If the efficiency of the dynamo is 90 per cent, and its output of energy is to be 134 H. P., the input or *delivered horse power* of the engine must be 134 ÷ 0.90 = 149.

If the efficiency of the engine is 85 per cent, the required I. H. P. will be 149 ÷ 0.85 = 175.

Putting this in the form of an equation we have,

$$\text{I. H. P. of engine} = \frac{\text{Kw.} \times 1{,}000}{746 \times A \times B}$$

in which

A = efficiency of engine.

B = efficiency of dynamo.

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Taking the lower efficiencies as given above, we have, I. H. P. of engine = Kw. of dynamo × 1.75 for dynamos up to 100 Kw., and Kw. × 1.57 for larger sizes.

The dimensions of different types and makes of machines will vary somewhat, and in making an accurate layout where space is limited, exact dimensions should always be obtained from the makers of the machines to be used.

The approximate floor dimensions of direct-connected sets may be obtained by adding the width of the engine-type generator to that of the required size of belted engine as given in Chapter XII. The total length, in most cases, will be the same as that of the engine, although in some instances it should be slightly increased.

In arranging dynamos, space should always be left for the removal of the armatures.

Installation

In designing a plant, it is customary for the engineer to first decide upon the type of engine best adapted to the needs of the case in hand, and then make a study of the mechanical construction of several standard makes fulfilling the requirements. This can be done from cuts and drawings furnished by the makers, supplemented, when possible, by an examination of similar engines already installed.

Foundations.—Detail drawings of the foundations are usually furnished by the builders of the engine. Care should be taken not only to make the foundations stable, but to construct them so that no vibration will be transmitted to adjoining rooms and buildings. A loose or sandy soil does not carry such vibrations as readily as a firm earth or rock. There are various ways of preventing the transmission of vibration, the most common of which is to make the excavation from 2 to 3 feet greater than the foundations in all directions, and fill in the space with well packed sand.

Both brick and concrete are used for the foundation itself and this is usually capped with hammered stone in the best class of work when bricks are used.

The anchor bolts are held in place during the construction of the foundation, by means of a wooden template which insures

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their correct position. They are commonly run inside of iron pipes with an internal diameter a little larger than the bolt; this allows a certain amount of play and is convenient for the final alignment of the machine. The lower ends of the bolts are provided with heavy iron plates or washers embedded in the masonry.

Generators are usually mounted on a wooden base to furnish insulation for the frame.

Data relating to the safe loading of different soils has already been given in connection with boiler and chimney foundations; and this applies in a general way to the foundations for engines and generators, although in this case vibration must be provided for in addition to the load itself.

For units of 200 Kw. and less it is customary to start with a footing of concrete 12 inches to 20 inches in thickness and of an area to be governed by the nature of the soil. If the foundation is to be of brick, it should have a batter of about 1 to 6. This, and the area of the footing will fix the depth.

Concrete is now generally used for the entire foundation in place of brick and stone. The general form and size in this case may be the same as already described.

Steam and Feed Piping

The design of a system of piping for a high-pressure steam plant is a matter of much importance, and should be carefully worked out in detail before any of the apparatus is installed. The boilers and engines and their auxiliaries should first be located on the drawings in their true positions, and then connected in a manner best suited to the existing conditions.

Piping Plans.—The plans for medium sized plants are usually laid out to a scale of $\frac{3}{8}$ or $\frac{1}{2}$ inch to the foot, and all valves and fittings drawn in their true proportion to make sure that there is sufficient space for the work to be made up. The principal points to be kept in mind when laying out a system of this kind are as follows: Pipes of sufficient size to prevent loss of pressure; reasonable compactness, to avoid excessive condensation; provision for expansion; freedom from pockets for the collection of water; and a system of drainage which shall keep the pipes free from condensation under all conditions.

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General Arrangement

In electric power plants of medium size, the arrangement of boilers and engines is much the same and usually follows one of two general schemes. In one case the boiler and engine rooms are placed end to end, with boilers and engines lying in the same direction, as shown in Fig. 168; while in the other arrangement they are placed back to back, with a wall between as indicated in

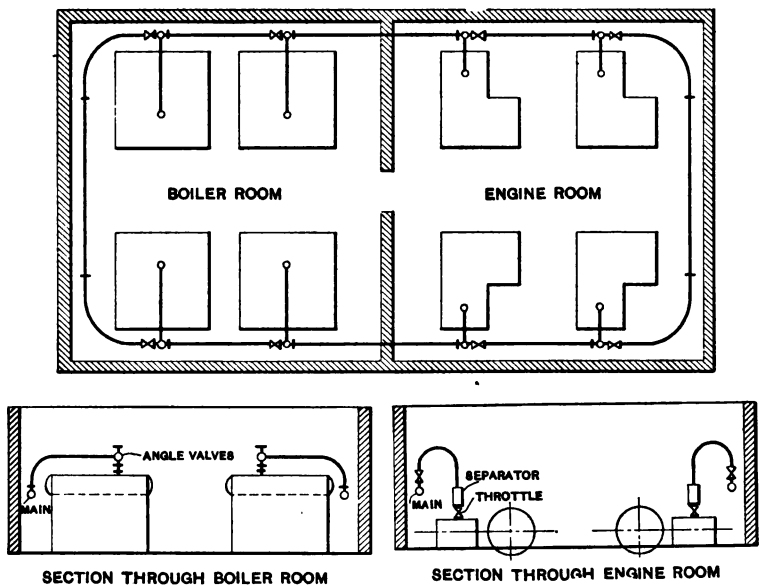

Fig. 168. Power Plant with Boiler and Engine Rooms Placed End to End.

Figs. 169 and 170. When the plant is located in the basement of an office building, or forms part of the power and heating system of some public building or institution, these arrangements cannot usually be carried out in full, and various modifications are necessary in each particular case. However, these diagrams will be found useful in many instances, even though the conditions are quite different, and will serve as a basis in laying out work of this kind.

High-Pressure Piping

Ring System of Piping.—The system of piping shown in Fig. 168 is known as the "ring" system. In this arrangement the steam

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passes from the boilers to the engines by two paths, and any section of the main may be cut out by closing two of the valves. The main, in this case, must be made full size for its entire length and the cut-out valves are necessarily of the same size, which adds considerably to the first cost. On the other hand, it practically serves the purpose of a duplicate system of piping, and in case of a leak or accident, the damaged section can be cut out by sacrificing a single engine or boiler, according to its location. Another arrangement of the ring system is shown in Fig. 169 where the boilers and engines are placed in much the same manner as in the back to back arrangement in Fig. 170.

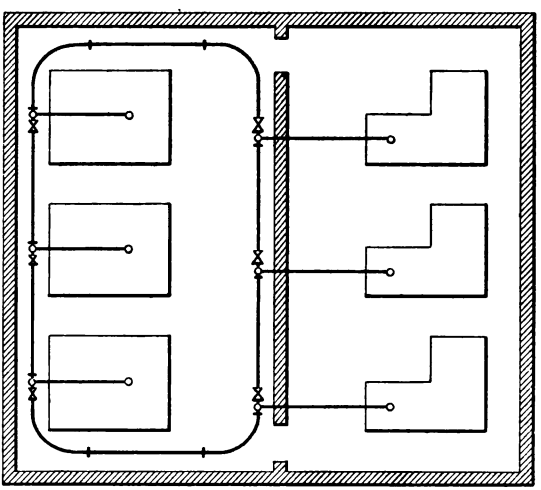

Fig. 169. Power Plant with Boiler and Engine Rooms Placed Back to Back.

Back to Back Arrangement.—In the back to back method the main or header is commonly carried on brackets secured to the boiler room wall. In this arrangement each boiler and engine has its valved connection with the header, which is common to all. In small plants this is usually made in one section, but where it is desired to guard against accidents so far as possible, it may be divided into sections as shown in Fig. 170. In this case only one boiler and engine will be thrown out of use by a break in the main, provided those connected with each section are properly proportioned. The main may be placed overhead as shown in Figs. 168

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and 170, or supported upon piers or brackets at the rear of the boiler settings.

The latter arrangement is commonly used in large plants where the engine room floor can be placed at a higher level than that of the boiler room. This allows the connections to be carried beneath the floor and does away with heavy exposed piping in the

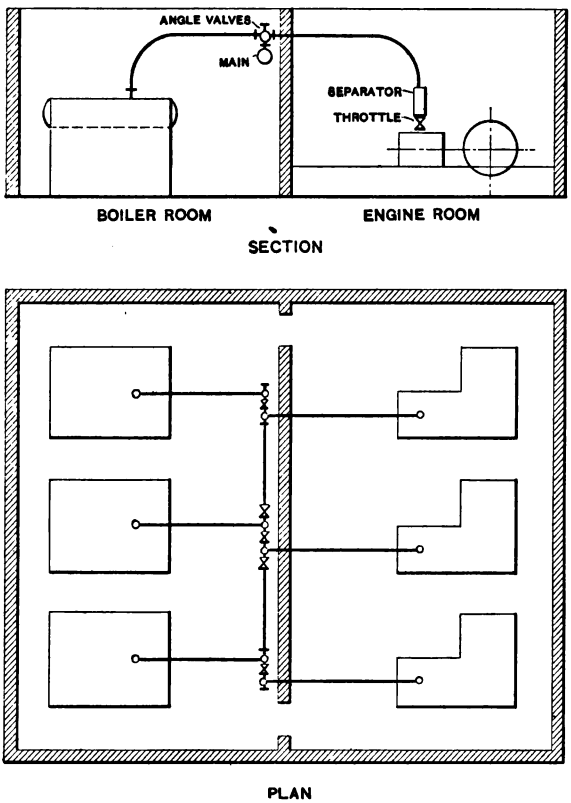

Fig. 170. Power Plant with Arrangement Similar to Fig. 169, but with Different Piping Plan

engine room. In this way the room can be kept cooler, and in case of leaks or breaks the escaping steam does not reach the electrical apparatus so readily. Although having these advantages, it can not usually be employed in office buildings and similar locations on account of the limited space.

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In making provision for expansion it is customary to use pipe bends and swivels so far as possible, in preference to slip joints.

Connections to the Boilers.—Common methods of connecting boilers with the main are shown in Figs. 168 and 170. In the first case an angle valve is placed directly upon the boiler nozzle, and the lead drops into the top of the main by means of a sweep bend. If head room is limited an elbow may be substituted for the angle valve, and a gate valve placed just beyond it, with the stem in a horizontal position. It is customary in the best class of work to place two valves in the lead, as a precaution against leakage of steam from the main into the boiler while the latter is shut down for inspection or repairs.

Connections to the Engine should always be taken from the top of the main, and if suitably located a full U bend may be used as shown in Fig. 168, with a gate valve in the pipe near the main, in addition to the throttle at the engine.

When there is sufficient head room a pipe bend is often connected directly to the boiler nozzle as shown in Fig. 170 and connection made with the top of the main either by means of an angle valve or an elbow and gate, the latter being placed in the horizontal portion of the pipe near the main.

The illustrations given show only a few of the methods in common use, and may be varied or combined in different cases to meet the requirements.

Cast-Iron Fittings.—In small plants, where moderate pressures are carried, cast-iron fittings may often be used in place of pipe bends, provided special care is taken to use swivel joints wherever possible. Rigid connections should be avoided in every case, and the engineer must always have in mind the flexibility of the piping under expansion and contraction.

Steam Separators are commonly placed close to the throttle of the engine. They not only serve to remove the moisture, but when of large size they provide a reservoir of steam close to the cylinder which insures a higher and more uniform pressure up to the point of cut-off, and also lessens the vibration in the steam pipe caused by the intermittent flow of steam to the cylinder.

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Exhaust Piping

The exhaust piping will vary considerably in different cases, depending upon whether the plant is to be run condensing or non-condensing, and also upon the disposal of the exhaust steam in non-condensing plants. When there is a sub-basement under the engine room, the exhaust piping may be carried in this space, together with the steam connections, otherwise they are run in covered trenches below the floor. The general method of making the exhaust connections under different conditions can best be illustrated by a few diagrams.

Exhaust Piping for Non-Condensing Plant.—Fig. 171 shows the method of connecting up a non-condensing plant in which the exhaust steam is used for heating purposes.

The exhaust main from the engines is carried beneath the floor and is so valved that the steam can be passed either outboard through a back-pressure valve, or into the heating system through a grease extractor. A closed feed-water heater is connected with the main so that either a part or the whole of the exhaust can be made to flow through it. This arrangement allows the heater to be cut out in case of repairs and also makes it possible to pass all of the exhaust through it in the summer time when the heating system is not in use, if so desired.

A cross connection is made with the high-pressure main for automatically supplying live steam to the heating system through a reducing valve in case the exhaust should prove insufficient at any time. The returns from the heating system are trapped into a vented receiver and pumped back to the boilers automatically when the waterline in the tank rises above a given point.

Connections to Feed-Water Heater.—Sometimes the feed-water heater has only a single steam connection as shown in Fig. 172. In this arrangement, which is sometimes called the "induction system," there is no circulation of steam through the heater, only enough being drawn in to take the place of that which is condensed. This condition makes it necessary to provide an automatic vent for the gases which are liberated from the steam, otherwise the heater would become air-bound.

When the heater is installed on the induction system it is necessary to place an independent grease extractor in the branch lead-

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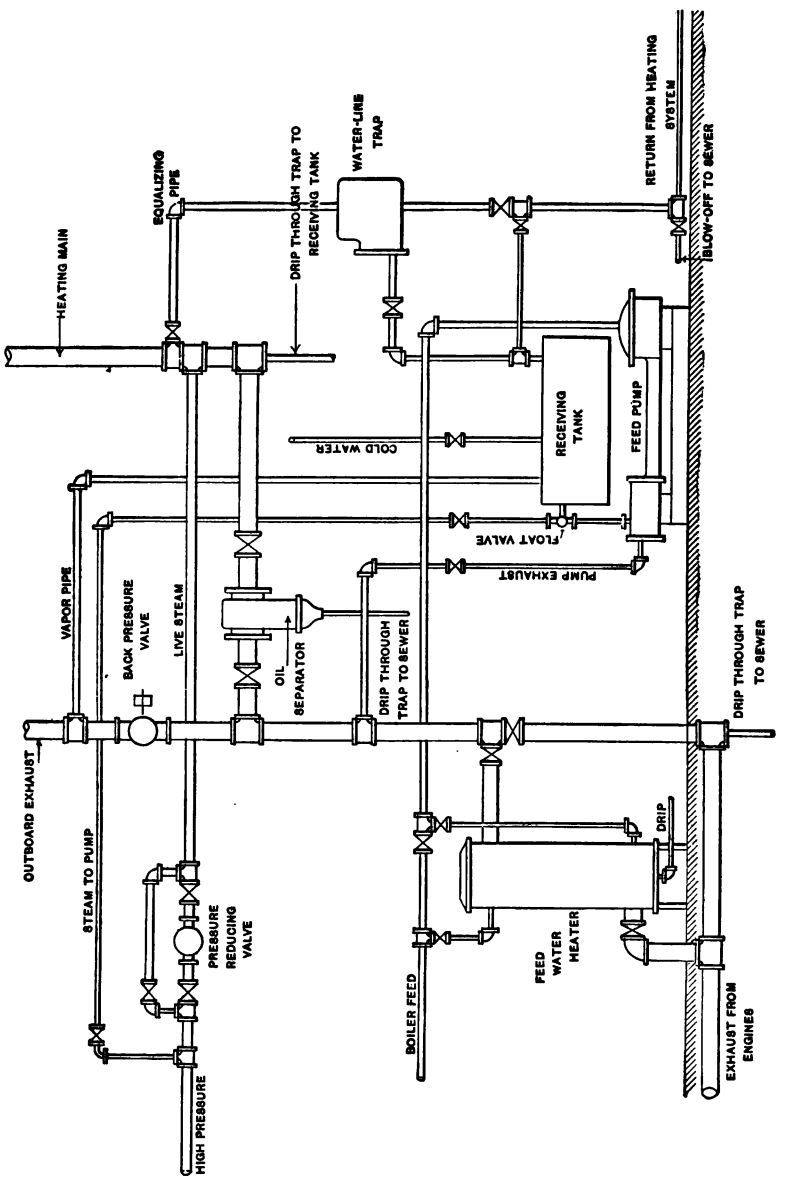

Fig. 127. Layout of Piping for Non-condensing Plant.

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ing to the heating system as shown. The separator should be drained through a special trap to the sewer or sump well.

This is a good method of connecting an open heater with an exhaust heating system, as it does away with any possibility of spray being carried over from the open pans or trays into the radiators, which might occur if the steam was passed through at too high a velocity.

Connections to Surface Condenser.—Fig. 173 shows the method of connecting with a surface condenser when the discharge water

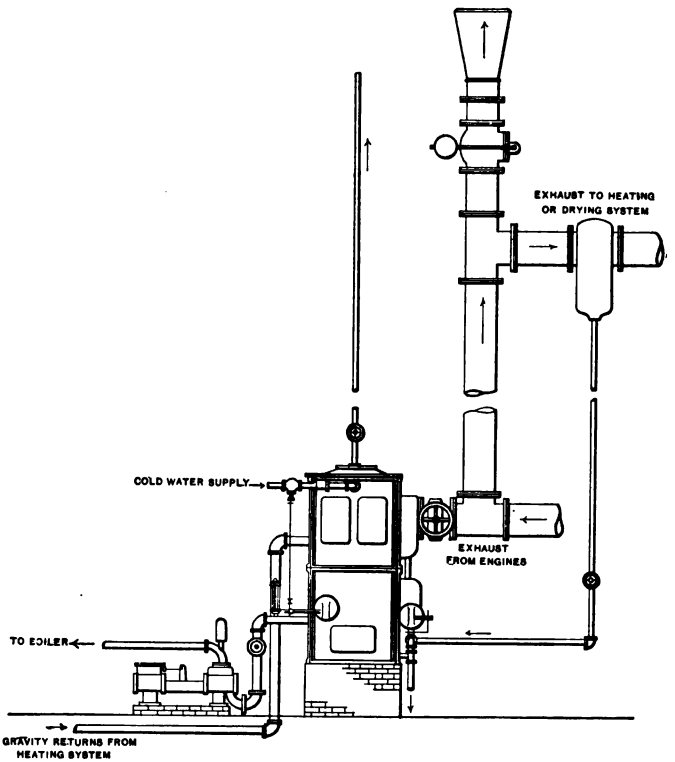

Fig. 172. Connections to Feed-water Heater.

is to be returned to the boilers. The piping is so arranged that the exhaust can be turned either outboard or passed into the condenser through an oil separator as desired. A relief valve is placed in the branch leading outboard which opens automatically

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in case anything happens to the condenser. The exhaust from the air and feed pumps should be turned into a heater.

It is evident that when in operation a partial vacuum exists in the separator as well as in the condenser, hence the drip cannot discharge against atmospheric pressure, and for this reason special provision must be made for drainage. One method of doing this is to lead the drip to a closed tank of good size having a float valve connected with a small steam whistle. When it becomes nearly filled with oil and water the alarm is given and the tank can be emptied by closing the drip pipe, opening the drain and admitting steam pressure to the top of the tank.

Another method which operates automatically is to use a good form of return trap in place of the tank. In this case, steam at low pressure should be connected with the trap for discharging its contents.

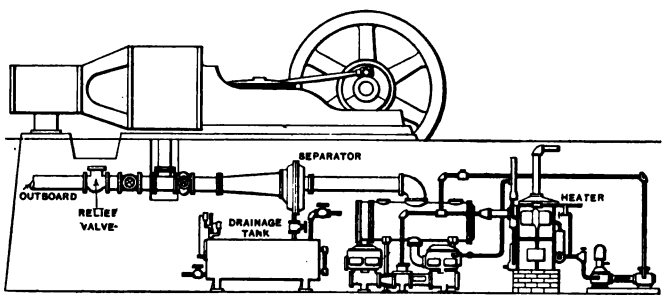

Fig. 173. Connections to Surface Condenser.

Connections to Jet Condenser.—Fig. 174 illustrates the connections for a jet condenser where the discharge water is turned into the sewer. In this case no oil separator is required, and a primary feed-water heater is substituted in its place. There is a connection outboard through a relief valve as before. The exhaust from the pumps should be discharged into a secondary heater through which the boiler feed should be made to pass after leaving the primary heater as already described in connection with feed-water heaters in a previous chapter. The arrangement is such in this case that all condensation from the heater is carried into the condenser with the flow of steam, so that no special drips are required.

Connecting to Siphon Condenser.—Fig. 175 shows the method

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of making the connections for a siphon condenser of the Knowles type. The condensing water may be taken from a tank or flume as shown, or may be supplied by a pump placed in the boiler or engine room.

Location of Condenser.—If it is not convenient to place the condenser at a lower level than the engine it may stand on the

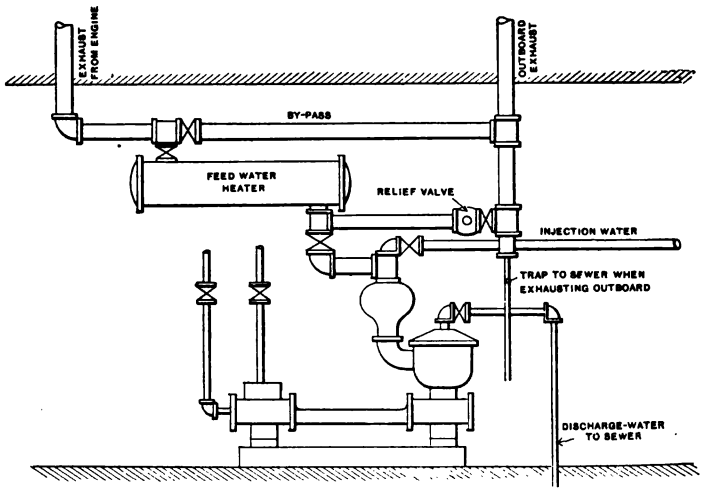

Fig. 174. Connections to Jet Condenser.

same grade; the only advantage in the arrangements shown being that of drainage.

In some plants each engine is provided with its own condenser, while in others, two or more are connected with the same one; in the latter case each engine should have a valve in the exhaust pipe near the cylinder.

Drainage

Of equal importance with flexibility is the matter of drainage.

Steam pipes should always be graded, if possible, so that the flow of steam and water will be in the same direction; if this cannot well be done the pipes should be made extra large. Pockets are to be avoided, but if for any reason they are found to be necessary in special cases, they should be effectively drained at the point where the pipe rises again.

Drips from Steam Mains.—When mains of considerable size

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are carried for long distances, they should be dripped at intervals of 125 to 150 feet into a line running back to the boiler room;

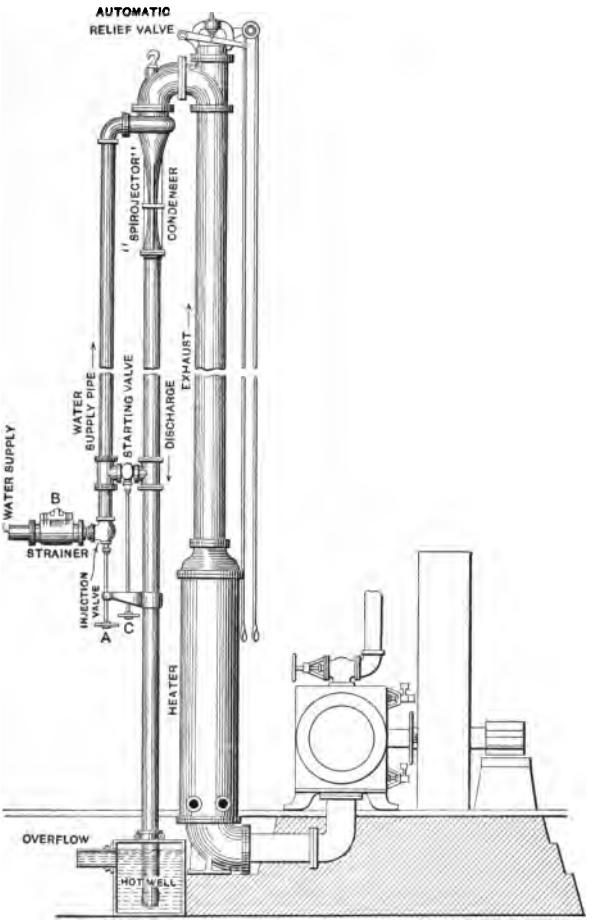

Fig. 175. Connections to Siphon Condenser.

and in case the main header is divided into sections, each should have its separate drip.

Location of Valves.—Globe valves ought never to be used in horizontal pipes with the stem in a vertical position, as a pocket will be formed for holding back the water of condensation; neither should a valve of any kind be used in a vertical pipe where

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it will form a pocket for the accumulation of water above it. This applies especially to vertical pipes above the nozzles of boilers.

Gate valves are better adapted for use in horizontal pipes under nearly all conditions, but care should be taken not to place them with the stem vertical and the hand-wheel at the bottom, lest they be left partially closed, and thus make an obstruction at the bottom of the pipe. When the throttle valve of an engine or pump is placed at the bottom of a vertical supply pipe, the pocket which is formed should be dripped by means of a "bleeder" connected into the pipe just above the valve. This may be blown out by hand just before starting up, or the drip may be connected with a trap which will keep the pipes clear automatically. The last arrangement is always necessary in the case of elevator pumps which run intermittently and are started automatically by a drop of pressure in the pressure tank.

Return of Condensation.—The condensation from piping containing live steam is usually collected and returned to the boilers by means of a pump or return trap. When the points to be dripped are under different pressures an automatic return pump with a vented receiver should be used, and each drip line provided with a trap discharging into it. If the drips are all under practically the same pressure and located near together, the condensation may be returned directly to the boilers by means of the "steam loop" or "Holly System," or they may be connected with the receiver of a return pump without the use of traps. If this is done, the different drip lines should be brought together below the water line of the tank and provided with check valves. This also applies to the use of a return trap.

Drips from Exhaust System.—The drips from oil separators, feed-water heaters, and all other parts of the exhaust system containing oil are usually trapped to the sewer. Cylinder cocks are commonly connected with the blow-off tank, which allows the steam to pass off through the vapor pipe.

Feed Piping

This is usually of brass in first-class plants, although extra heavy iron pipe is sometimes used. Long-turn fittings and gate valves are to be preferred in this class of work.

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The method of connecting up the feed apparatus will vary in different cases, but the following diagrams will serve as a basis from which to make up different combinations as may be required.

For simplicity only one piece of apparatus of a kind is shown

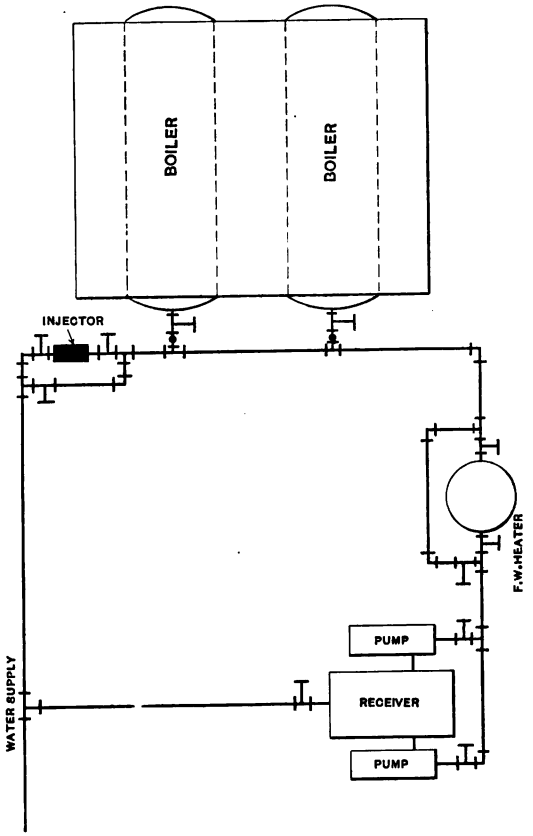

Fig. 176. Feed Connections for Non-condensing Plant.

except in the case of feed pumps, which should always be in duplicate.

Feed Piping for Non-Condensing Plant.—Fig. 176 shows the feed connections for a non-condensing plant of the simplest kind. The supply main separates into two branches, one leading to the receiving tank and the other to an injector near the front of the

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boilers. The connection with the tank may have simply a hand valve for supplying fresh water when required, or it may be provided with a float valve for admitting it automatically when the water line in the tank falls below a given level. The discharge

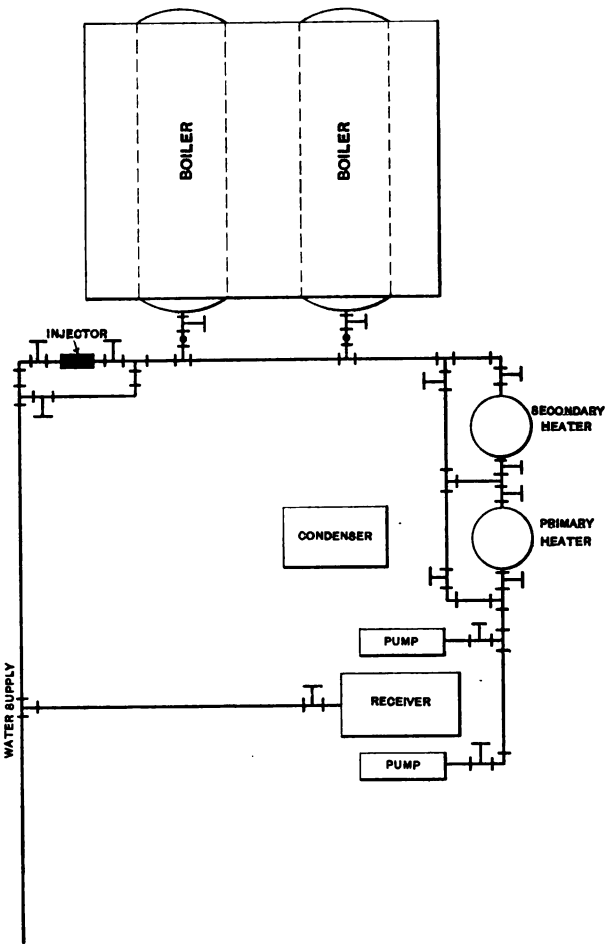

Fig. 177. Feed Connections for Plant with Jet Condenser.

from the feed pump is passed through a closed heater on its way to the boilers. The branch leading to each boiler should be provided with both check and gate valves, the latter being placed next to the boiler and in a position easily reached from the boiler

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room floor. A by-pass is shown around the injector for admitting water to the boilers directly from the main.

Sometimes the injector is placed on a separate line; this is a good arrangement in large plants, as it admits of any boiler being

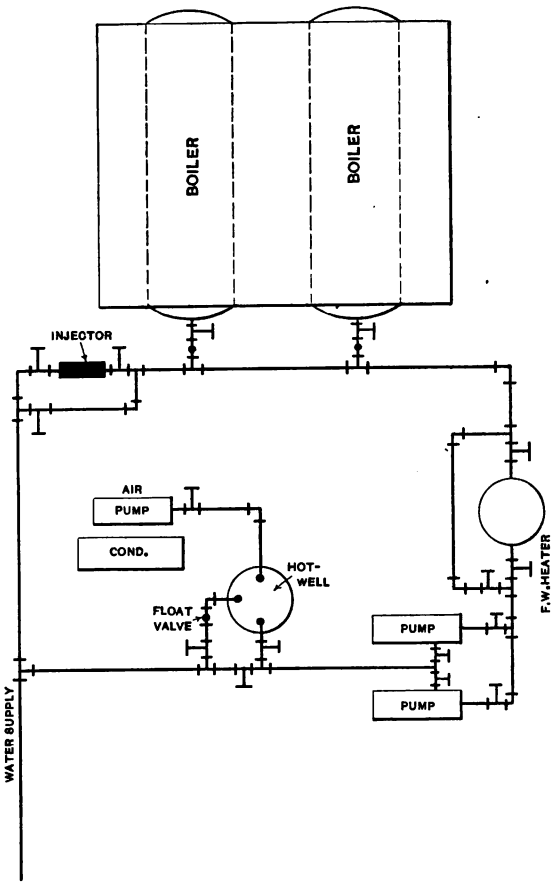

Fig. 178. Feed Connections for Plant with Surface Condenser.

emptied and refilled from the main, independently of those under steam pressure. All drips, and the returns from the heating system are trapped into the receiving tank. An open heater might be used in this case which would take the place of both the receiving tank and closed heater.

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Feed Piping for Condensing Plants.—Fig. 177 shows the arrangement of piping for a plant using a jet condenser in which the discharge water is turned into the sewer. The only change in this case is the addition of a primary heater placed in the main exhaust pipe between the engine and condenser. The discharge from the feed pump is first passed through this heater and then through an auxiliary supplied with exhaust steam from the air and feed pumps.

Fig. 178 illustrates a condensing plant in which a surface condenser is used. The exhaust from the engine is passed through an oil separator and the discharge water from the air-pump turned into a hot-well, from which it is pumped back to the boilers. A feed-water heater taking exhaust steam from the pumps, is used as before, but the fresh water supply, high-pressure drips, and return from the heating system are discharged into the hot-well instead of a receiver. The pump suction has a cross-connection so that it can take its supply either from the hot-well or directly from the main.

In case an economizer was used, supplying heat from the flue gases, it should be connected into the feed line between the exhaust heater and the boilers.

Sizes of Steam Pipes

The sizes of steam pipes for high-pressure work, where the length of run does not exceed 100 feet, are usually based on the velocity of flow. The maximum commonly allowed is 6,000 feet per minute for supply mains and 4,000 feet for exhaust. In practice it is customary to make the pipes slightly larger than called for by this velocity.

Computing Size of Piping.—The following empirical formulas may be used for computing the sizes of steam and exhaust pipes of modern engines of good economy, when the cylinder dimensions and speed are known:

For steam pipe,

$$d = D\sqrt{\frac{L \times R}{36{,}000}}$$

For exhaust pipe,

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$$d = D\sqrt{\frac{L \times R}{24{,}000}}$$

in which

d = diameter of pipe in inches.

D = diameter of cylinder in inches.

L = length of stroke in inches.

R = revolutions per minute.

A simple method giving a slightly larger pipe than the above, but corresponding well with average modern practice, is to allow 0.12 square inches area per I. H. P. for steam, and 0.18 square inches for exhaust pipe. The factor 0.12 may also be applied with good results to boilers, as 30 pounds of steam per H. P. per hour at 70 pounds pressure represents the average water rate of simple non-condensing engines as well as the quantity furnished by one boiler H. P.

Example.—A power plant consists of six 100 H. P. boilers which supply steam to four 150 H. P. engines. All of the boilers are to be connected with a main drum or header, from which supply pipes are to be carried to each engine. What should be the size of the boiler leads, the main header, and the supply and exhaust connections with the engines?

Size of boiler leads equals $100 \times 0.12 = 12$ sq. in., which corresponds very closely to a 4-in pipe. The main header is commonly given a sectional area equal to that of all the leads connecting with it, which in this case would call for $600 \times 0.12 = 72$ sq. in., which is slightly less than a 10-in. pipe. Size of steam supplies to engines equals $150 \times 0.12 = 18$ sq. in.; this is somewhat less than the area of a 5-in. pipe, but as $4\frac{1}{2}$ in. is not a standard size, the former would probably be used. The exhaust connections call for $150 \times 0.18 = 27$ sq. in., which corresponds to a 6-in. pipe. In practice, however, pipe sizes are usually fixed by the outlets provided on boilers and engines.

Steam Main with Reservoir near Engine.—In some of the latest high-pressure plants, the pipes leading from the main header to the engines have been made from one to two sizes smaller than called for by the engine builders. These pipes connect with large separators having a cubic capacity about three times that of the high-pressure cylinder and placed close to the throttle valve of the engine. The connection between the separator and cylinder is

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made the full size called for by the engine. The object of this arrangement is to have a full supply of steam close to the throttle; to provide a cushion near the engine on which the blow caused by the cut-off may be spent, thereby preventing vibrations from being transmitted through the piping system; and, finally to produce a steady and rapid flow of steam in one direction, by having a small pipe leading to the receiver.

Sizes of Feed Pipes

Feed Pipe.—The size of feed pipe is usually governed by the discharge outlet of the pump. Special makes of boilers, feed-water heaters, etc., have the size of feed connections fixed, so that in piping up the different pieces of apparatus it is merely a matter of proportioning the pipes to the openings to be connected.

In the case of tubular boilers the size of feed pipe is usually fixed by custom, and depends upon the diameter of the boiler. The dimensions in Table LXV. may be taken as about the average practice.

TABLE LXV.

SIZES OF FEED PIPE.

Diameter of boiler, inches	Size of feed pipe, inches
42	1
48	1
54	1¼
60	1¼
66	1¼
72	1½

Proportioning Pipe Sizes.—The common method of proportioning pipe sizes is to make the total sectional area of the branches equal to that of the main, although as a matter of fact it should be somewhat greater. This is true because for a given area and velocity of flow, there will be less frictional resistance in a single large pipe than if the same quantity of water were passed through several smaller pipes having the same total area.

Table LXVI. gives the internal area in square inches of several sizes of standard wrought-iron pipe.

Table LXVII. takes into account the frictional resistance as noted above and shows the number and diameter of branches which a given main will supply. This is similar to Table LXIII., but includes a number of smaller sizes.

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TABLE LXVI.

INTERNAL AREA OF STANDARD WROUGHT IRON PIPE.

Diameter of pipe, inches	Internal area, square inches	Diameter of pipe, inches	Internal area, square inches
¾	0.53	3	7.4
1	0.86	3½	9.9
1¼	1.5	4	12.7
1½	2.0	5	20.0
2	3.3	6	28.9
2½	4.8	8	50.0

TABLE LXVII.

NUMBER AND SIZE OF BRANCH PIPES WHICH A GIVEN MAIN WILL SUPPLY.

Main	Branch
1 in. will supply	two ..... ¾ in.
1¼ in. "	two ..... 1 in.
1½ in. "	two ..... 1¼ in.
2 in. "	two ..... 1½ in.
2½ in. "	two 1½ in. and one 1¼ in., or one 2 in. and one 1½ in.
3 in. "	one 2½ in. and one 2 in., or two 2 in. and one 1½ in.
3½ in. "	two 2½ in. or one 3 in., and one 2 in. or three 2 in.
4 in. "	one 3½ in. and one 2½ in., or two 3 in. and four 2 in.
4½ in. "	one 3½ in. and one 3 in., or one 4 in. and one 2½ in.
5 in. "	one 4 in. and one 3 in., or one 4½ in. and one 2½ in.
6 in. "	two 4 in. and one 3 in., or four 3 in. or ten 2 in.
7 in. "	one 6 in. and one 4 in., or three 4 in. and one 2 in.
8 in. "	two 6 in. and one 5 in., or five 4 in. and two 2 in.

Insulation

Steam pipes, separators, heaters, etc., should be covered with some good insulating material to reduce the amount of condensation, and incidentally to prevent, to some extent, overheating the engine and boiler rooms.

Materials Used.—Various materials are used for this purpose, the more common being carbonate of magnesia, asbestos, hair-felt, cork and mineral wool.

These are made up in various combinations and formed into canvas-covered sections about 3 feet in length, as shown in Fig. 179. After being placed upon the pipe, the canvas flaps are pasted down and light metal bands drawn tightly around the ends and middle of each section to hold it in place.

Valves and fittings are covered either with moulded or plastic material. The former consists of a mould of proper size and

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shape to enclose the fitting, made in two parts and held in place by a canvas covering and metal bands. One of these is shown in Fig. 180.

Plastic Covering is made by applying several thin coats of asbestos or magnesia in the form of a paste directly to the fitting, each being allowed to dry before the next is applied. This is

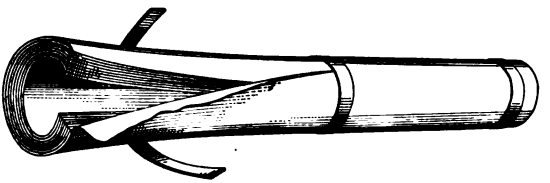
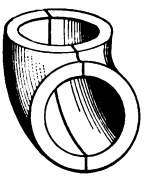
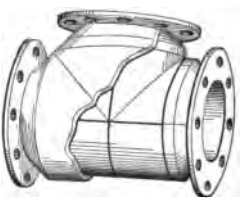


Fig. 179. Pipe Covering.

given the proper shape, and after hardening is covered with canvas securely pasted to hold it in place.

Extended surfaces, like small boilers, heaters, smoke pipes, etc., are usually covered with plastic material applied to a wire netting secured to the metal surface. A smooth finish is given when nearly dry by rubbing with a trowel.





Figs. 180 and 181. Covering for Fittings.

Sometimes a lagging of insulating blocks an inch or so in thickness is first wired to the metal surface, and this finished with a coat of plastic material. Fig. 181 shows this method of insulation applied to a large fitting.

Pipe coverings are usually made from 1 inch to 1¼ inches in thickness, this being the economical limit; any additional heat saved by using a thicker covering being slight when compared with the increased cost.

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Plastic covering possesses the advantage of being applicable to any shape of pipe or fitting, as well as boilers, heaters, separators, etc., but the covering cannot be removed for the inspection of joints and seams without being destroyed. Sectional covering can be taken off and replaced without damage, and is used almost exclusively for straight runs of pipe. Moulded fittings are commonly used in the best class of high-pressure work, especially for flanges, while a plastic covering is usually sufficient in the case of low-pressure heating systems.

Selection of a Covering.—In selecting a covering, its durability should be considered as of equal importance with its non-conducting properties. Many coverings which show a good efficiency when new, deteriorate quite rapidly when placed upon a pipe line connected with high-speed engines or other machinery causing vibration.

Coverings containing corrosive substances, such as sulphate of lime (plaster of Paris) and sulphur should always be avoided. Carbonate of lime (chalk) is harmless so far as corrosion is concerned, but is reduced to a powder by vibration and thus rendered useless. Some of the inferior grades of mineral wool are apt to contain sulphur, which is likely to attack the pipe if it becomes moist.

Carbonate of magnesia and asbestos fiber both make durable and efficient coverings when properly made up, and in specifying, the words "carbonate" and "fiber" should always be mentioned when referring to these materials.

Cork makes a good covering when properly prepared. The granulated material is subjected to a high pressure in iron moulds at a temperature of about 500°. It commonly has a thin layer of asbestos next to the pipe and is provided with a canvas cover the same as other forms. This is especially adapted to places exposed to dampness, as the cork is impervious to moisture.

Heat Loss from Pipes.—The heat loss from bare pipes varies from 2½ to 3 B. T. U. per square foot of surface per hour, for each degree difference in temperature between the steam and surrounding air, depending upon the size of pipe and the freedom with which the air circulates about it. A series of tests made some years ago at the Massachusetts Institute of Technology upon a dozen or more commercial coverings showed that from

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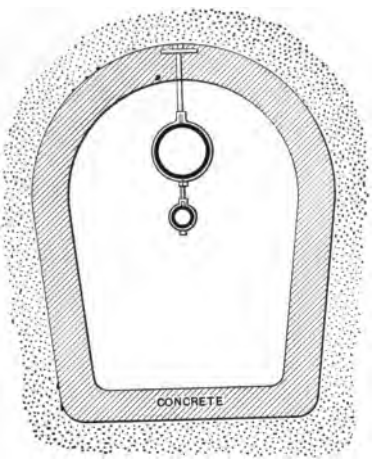
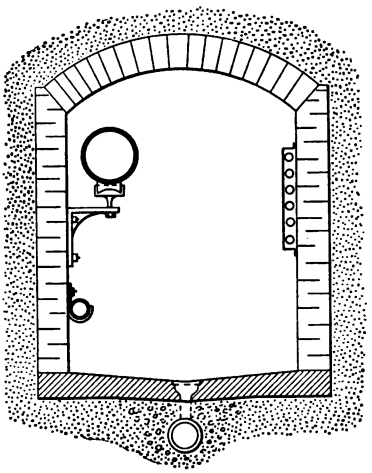
75 to 84 per cent of the heat lost from the bare pipe was saved. Taking the average as 80 per cent, and assuming a loss of 3 B. T. U. per square foot per degree difference per hour, the saving to be made in any particular case can be easily approximated.

Underground Conduits

There are various forms of conduits employed for carrying pipes underground where isolated buildings are supplied with steam and electricity from a central plant.





Figs. 182 and 183. Pipe Conduits.

Arched Conduit.—The best type will depend upon circumstances. For high-pressure mains of large size, where the distance is not too great, an arched tunnel of brick or concrete is the best. This should be large enough for a person to pass through easily for purposes of inspection or for repairs. The pipe may be supported from hangers attached to plates embedded in the masonry of the arch, or adjustable brackets may be used, anchor-bolted to the side wall, as shown in Figs. 182 and 183. Mains carried in tunnels of this kind should be protected with a good form of sectional covering, coated with asphaltum or a good waterproof paint. Pipes of smaller size are commonly carried in closed conduits of tile, cement, or wood.

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Storage and Handling of Coal

The storage of coal, and its method of conveyance to the boiler furnaces are items of much importance and should receive careful attention when designing a power plant, even of small size.

The amount of storage space necessary will depend largely upon the regularity of fuel delivery at the plant under all circumstances.

When possible, bins should be provided so that the coal may be delivered through chutes either onto the floor in front of the boilers or into the hoppers, if stokers are used.

When the boilers are in the basement or sub-basement of a city building, the coal can often be dumped from wagons directly into the storage bins through suitable chutes, and from here delivered by gravity to the boiler room. If the available space in the building itself is limited, provision is often made outside for the general storage of coal, and means provided for conveying it economically to the bins in the boiler house as needed.

Conveyors.—There are various forms of elevators and mechanical conveyors by means of which coal may be transferred from cars or wagons to the storage bunkers, and from there to the furnaces, if the bunkers are located outside the building. Conveyors of this kind are often arranged to pass beneath the ash hoppers under the furnaces, and to deliver the ashes to elevated bins, from which they may be discharged into wagons or cars for removal from the premises.

A good idea of the various arrangements for the handling of coal and ashes by mechanical means may be obtained by an examination of the illustrated catalogues of manufacturers of this type of equipment.

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CHAPTER XVI.

CARE AND MANAGEMENT OF STEAM BOILERS AND ENGINES

The best methods to be employed in the management of any particular boiler plant can only be learned by experience. There are, however, certain rules and precautions which apply in a general way to all power plants and a brief enumeration of these may be found of value to the designing engineer when starting up a new plant.

Boilers

Water Level.—Before starting up the fires under a battery of boilers the fireman should first determine the water level in each. This should be done both by an inspection of the water-glasses and by trying the gauge cocks. The former is a quicker and more convenient method but may be misleading on account of the presence of dirt and foam. On the other hand, when the steam goes down in a boiler which is perfectly tight, a vacuum is formed which will prevent the water from appearing when the cock is opened, although the boiler may have a sufficient supply and the glass indicate the true level.

From this it is evident that neither the water glass nor the gauge cocks should be depended upon alone, but used to supplement each other. When a vacuum exists in the steam space as above described, the cocks should be opened and air admitted until the water flows from them freely before starting the fire. The admission of air will take place most rapidly through the upper cocks, provided they are above the water line; if not, the safety valve may be opened slightly.

After starting the fire, the gauge cocks or other outlet should be left open until steam appears. This allows the air to escape again as steam is formed to fill its place. After making sure of the true water line, the safety valve should be raised slightly, if it has not already been done, to see that it is in working order.

Starting the Fire.—The method of starting a fire in a boiler furnace will vary somewhat with the kind of fuel used.

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In the case of bituminous coal a wood fire is first kindled directly on the grate, the sticks being so arranged as to allow the air to pass freely beneath them. A light layer of coal is added, then with the furnace doors closed and the draft doors open, the fire is allowed to burn until the coal has become thoroughly ignited.

It is now evenly distributed over the entire grate and fresh fuel supplied as may be required. When using anthracite, the method is practically the same, except a thin layer of coal is first spread over the grate before putting on the wood. When starting a fire under a cold boiler, it should be allowed to burn slowly at first so that all parts of the boiler and setting may be warmed up as evenly as possible, in order to avoid unequal expansion.

In the case of water-tube boilers the fire may be forced more at the beginning if necessary, owing to the smaller amount of water in the boiler and the greater distance of the joints from the hottest part of the fire. After the fire is once started, the method of adding more fuel depends upon the type of boiler, the kind of coal used, and the rate of combustion.

Methods of Firing.—There are three general methods of firing, known as "alternate" or "side" firing, "spreading," and "coking." In the first of these, fuel is supplied alternately to the opposite sides of the furnace. This has the advantage of a minimum cooling effect, and also provides sufficient heat for burning the gases which are distilled from the freshly fired coal, as they are in close contact with the incandescent fuel upon the other side of the grate. Moderate amounts of coal are fired at regular intervals and fresh air admitted to the furnace after each by means of the draft plates.

In spreading, as the name implies, small quantities of coal are fired at short intervals over the entire surface of the grate. One objection to this method is the frequent opening of the fire doors, which tends to cool down the furnace. To get the best results, a somewhat thicker fire should be carried at the sides than at the center, and air should be admitted for a short time after firing fresh coal the same as in the alternate method. Both of these systems are best adapted to the burning of anthracite coal on account of its freedom from the hydrocarbons.

Bituminous coal gives the best results when the coking method

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is employed, and a number of special furnaces have already been described, designed particularly for this method of firing. Although the construction of these may vary somewhat, the principles involved and the results sought, are practically the same in each.

When the coking method is employed in the ordinary hand-fired furnace, the fresh coal is placed just inside the feed door and allowed to remain from 20 to 30 minutes, while the hydro-carbons are driven off by the heat. During this period air is supplied for their combustion by opening the draft plates. When the coking process is practically completed, the unburned portion is pushed back over the fuel bed where combustion is completed with the burning of the carbon. The best thickness of fire must be determined by trial and will depend upon the grade of coal used and the available draft. In case of a strong draft and coarse coal the fire may be about a foot in thickness, but if the draft is weak or the coal fine, the thickness should not be more than four or five inches.

With forced draft, a thicker bed of coal must be carried in order to secure a high rate of combustion. With the same draft, bituminous coal may be fired more thickly than anthracite. After having determined by experiment the best thickness of fire to carry, it should always be kept the same. The rear of the grate should never be allowed to get bare, and air holes in the fire should be covered over as soon as formed. If it becomes necessary to force a boiler it may be done by firing smaller quantities of coal at shorter intervals.

Cleaning the Fire.—There are two common methods of doing this, one being to clean one side at a time, and the other to first clean the front and then the back. In some cases these are combined to some extent, different firemen having their own special ways of doing it. Whatever method is employed, the fire should be allowed to burn down somewhat, and care be taken to start with plenty of water in the boiler to last through the process. The work should be done as rapidly as possible, in order not to deaden the fire to such an extent that it will not come up quickly afterward.

Banking the Fire.—This is usually done by first cleaning and then pushing the coals back toward the bridge wall and covering

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with fresh fuel. The fire doors should be left open, and the damper closed, except as may be needed to carry off the small amount of gas formed. When starting up a fire which has been banked, the same precautions should be noted as when building a new one.

Blowing off.—The frequency with which this should be done will depend somewhat upon local conditions. If the water is muddy or contains oil, a few inches should be blown out daily, and the boiler should be emptied every week or ten days, and oftener if the water is very impure. This should never be done, however, while the boiler is hot. Certain salts of lime and magnesia, frequently found in feed water, are precipitated when heated above a certain temperature, and should be blown out daily, under a low pressure. A good time to do this is in the morning before starting up the fires, or at night after they have been allowed to die down somewhat. When the water ceases to boil, much of the sediment, which at other times is carried back and forth through the boiler by circulation is deposited, usually in the coolest part, and a large part of it may then be blown out. Blowing off when there is a hot fire in the furnace does but little good unless the boiler is provided with a mud drum which is protected from the heat of the fire.

When the water contains oil or organic impurities which float on top, the surface blow-off should be used.

Sudden Shut-Down.—If it becomes necessary to shut down the engine with a heavy fire under the boiler and steam at the blowing point, the drafts should be closed and the pump or injector started to keep the pressure from rising. The gauge should be closely watched until the safety valve lifts, to see that the latter works freely. If the safety valve is of sufficient size and in working order, the pressure cannot rise dangerously high with the pump or injector forcing cool water into the boiler.

Low Water.—In case of low water, cover the fire with wet ashes, or fresh coal, if the ashes are not at hand. Do not draw the fire until it has become so deadened that it can be done without increasing the heat. The safety valve should not be lifted at such a time, neither should cold water be fed into the boiler. The engine should be allowed to run until it stops, when the throttle valve may be closed. All conditions should be kept as uniform as

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possible until the fires can be checked and the boiler cooled down. In case of a fracture in the main steam pipe the water level is usually lowered very rapidly, and the first thing is to deaden the fires. If the rush of steam is not too great, and the gauge glass can be kept in view, the water level may often be maintained by starting or increasing the speed of the pump.

Sudden leaks in the blow-off pipe or the melting of a fusible plug call for the same procedure as above.

Prevention of Foaming and Priming.—Foaming is usually caused by dirty or oily water, in which case the remedy is to blow off the boiler and add fresh water.

It may sometimes be stopped temporarily by closing the valve in the steam pipe for a short time until a normal condition is obtained within the boiler.

Priming is generally caused by carrying the water level too high or by forcing the boiler too far beyond its normal capacity. The remedy for this condition from either cause is evident.

Safety Valve and Pressure Gauge.—The safety valve should always be provided with means for lifting it, accessible from the boiler room floor, and should be tested daily.

The pressure gauge should be compared with a standard gauge occasionally in order to check its accuracy. If at any time the pressure should rise above the range indicated by the gauge, and the safety valve fail to blow, the fire should be checked as quickly as possible, and a careful examination made of both gauge and valve.

Purification of Feed Water.—A careful study of this subject based upon the experience of a large number of designing and operating engineers seems to indicate that the only satisfactory method of dealing with this important matter is to submit a sample of the feed water to a competent chemist for a careful analysis, and to obtain from him a suitable remedy for neutralizing the injurious effects of the impurities found. The leading boiler insurance companies do this free of charge for their clients. The great variety of impurities, and the varying quantities of each found in different localities, make it impossible to give definite solvent or precipitating solutions suitable for general use.

Water Softening Plants.—Although the various methods of purifying the feed water may be carried on to a large extent in

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the boilers themselves, it is now quite general in the case of large power plants to employ special apparatus for this purpose.

This method of purifying and softening the water does away to a large extent with the washing and cleaning of the boilers, except at long intervals, and results in a considerable saving in labor and repairs. Another saving is in the heat lost in the frequent blowing off of boilers for the removal of scale and mud.

In a general way the purification of feed water by this method consists in adding certain solutions of lime and soda and passing through suitable settling tanks and filters before feeding to the boilers.

Inspection and Cleaning.—Boilers should be examined frequently, both inside and out, for any appearance of deterioration.

Flues and heating surfaces should be kept free from ashes and soot, the frequency of cleaning depending to some extent upon the kind of fuel. Wire brushes, scrapers, and steam and air blasts are used for this purpose, depending upon the type of boiler. While the formation of scale upon the interior surfaces of a boiler should be reduced to a minimum, it is not always possible to overcome it entirely, and means must be taken for removing it.

This is done mechanically, and also by the use of solvents which tend to soften or loosen the scale so that it may be removed more easily. A safe method, but one which requires a good deal of time, is to chip it off with suitable tools. This, however, becomes difficult when the scale is hard or the construction of the boiler is complicated.

There are various forms of mechanical cleaners in use, designed both for water-tube and fire-tube boilers. The turbine cleaner which is used for removing scale from the inside of the tubes consists of a revolving head, carrying a number of small toothed wheels for cutting the scale. The cutters are usually driven by a small water motor so arranged that the discharge passes into the tube and serves to wash away the chips from the cutters. When the scale is on the outside of the tube as in the case of a fire-tube boiler, a similar arrangement is used in which the cutters are replaced by small weights or hammers which strike against the inside of the tube in rapid succession and produce a vibration which is sufficient to loosen the scale from the outside.

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Kerosene oil is used to some extent for the prevention and removal of scale. When used in the proper quantity it breaks up and loosens hard scale and prevents its formation. It is best introduced drop by drop by means of an oil feeder, and only enough should be used to produce the desired result, as an excess is likely to be carried over in the steam and to injure the interior surfaces of the engine cylinders.

Boiler Standing Idle.—When a boiler is to be laid off for some time precautions must be taken against the formation of rust on the interior surfaces. There are different ways of doing this, the most common being to fill it with a weak solution of soda, and the other to empty and thoroughly dry it, then place pans of quick lime in the bottom to absorb any moisture which may find its way in.

Engines

Although a properly designed and well made engine requires but little care while running, the engineer in charge must always be on the alert for emergencies and keep a careful lookout for any indication of defective operation.

The principal matters needing attention are lubrication and proper adjustment of the bearings and valves.

Lubrication.—The first item of importance is the selection of a suitable oil. For cylinder lubrication a heavy mineral oil should be used, as animal oils are decomposed by heat and form a hard coating upon the internal surfaces of the cylinder and steam chest. For journals and other bearing surfaces this should be thinned somewhat with a lighter organic oil. Cylinder lubrication is usually effected by means of an automatic sight-feed lubricator, in which the oil is made to pass drop by drop through a glass tube filled with water, as it is fed into the steam pipe above the throttle valve. The method of supplying oil to the different bearings varies with the make and type of engine. For slow-speed engines, oil cups are commonly used, but with high-speed engines it is customary to use some form of forced lubrication, either by means of a small pump or by feeding from an elevated reservoir.

The main bearings are often provided with a tallow cup in the cap of the bearing, in addition to the regular oil cup or auto-

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matic feed. This is for use in case the oil supply fails for any reason and the bearing becomes heated, as under these conditions the tallow will melt and flow into the bearing as needed. Should it ever become necessary to cool a bearing with water, it must be done very slowly in order to prevent cracking or warping the metal by too sudden contraction.

Adjustment of Bearings, Stuffing Boxes, etc.—This is an important matter which requires constant attention. Horizontal engines should be watched for signs of wear upon the lower part of the cylinder, due to the weight of the piston. The piston rings require careful adjustment in order to avoid cutting the cylinder walls, and the piston rod must be kept exactly central to prevent excessive wear in the stuffing box and also to prevent springing the rod itself.

Stuffing boxes must be kept well packed and adjusted, and the packing should be renewed as often as necessary to keep a tight and smoothly working joint.

The connecting-rod bearings at both the crank and wrist pins should receive careful attention, as the "knocking" so often heard in engines is generally caused by a slight wearing of the brasses at these points.

The main bearings should be examined frequently, and any lateral movement taken up by proper adjustment of the side boxes, and in the case of high-speed engines the caps should be tightened occasionally to prevent any upward movement of the shaft.

Starting an Engine.—Before starting an engine the condition of the boilers should be noted to see that steam is up and the water-line at its normal position. All tools and loose parts must be removed from the engine, oil cups and cylinder lubricator filled and adjusted, and the drain cocks opened. The throttle valve may now be opened sufficiently to allow a small quantity of steam to blow through to warm up the cylinder and adjacent parts and to clear them of condensation. In engines of large size it is customary to work the valve gear by hand for a few revolutions until it is seen that everything is in running order. Those of smaller size may be slowly brought to speed by means of the throttle valve alone. The drip cocks should remain open until all water is expelled from the cylinder.

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An engine should be started slowly, not only as a precaution against accidents to itself, but also to avoid priming at the boilers, which is liable to occur if a sudden rush of steam is produced through the nozzles. When a condenser is used it should be started first, and stopped only after the engine has been shut down. In the case of a surface condenser, first start the air pump and when a sufficient vacuum has been formed start the circulating pump and then the engine as already described. If the condenser has a combination air and circulating pump, they must of course be started at the same time.

The grease which accumulates on the tubes of a surface condenser should be removed occasionally by the use of a strong solution of soda. If a jet condenser is used, first start the pump, then partially open the injection valve and run the pump rapidly until the condensing water comes. Then speed the pump so that it will be just ahead of the water and the water valves can be heard to close with force. This will insure a proper emptying of the condenser and cause the highest vacuum to be formed. The engine may now be started as already described.

Priming.—This usually makes its presence known by a "slapping" sound in the cylinder, which is produced by the piston striking the water and throwing it against the cylinder heads. As soon as this sound is heard, the cylinder drains should be opened and a hurried inspection made of the separator to see if it is working properly. If there is considerable water in the glass on the separator, the by-pass around the trap which drains it should be opened wide.

When these precautions have been taken in the engine room, the engineer should visit the boilers and order the firemen to cover the fires of the priming boiler with fresh fuel. Meanwhile the water-level in the gauge glasses should be noted, and if any of them show high water the blow-off should be opened and the water brought down to its proper level. These precautions will, in nearly every case, stop the priming; but if they do not, the stop valve on the offending boiler should be partially closed so as to wire draw the steam and thus dry it. The engineer should always have a general oversight of the boilers, and before shutting down the engine he should notify the firemen so that the fires may be allowed to burn down somewhat before this is done.

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Steam Turbines

In general, the same precautions noted in connection with a steam engine apply to a turbine of the same size.

Particular care should be exercised in starting up a machine to thoroughly warm up all parts to their proper temperature, to prevent distortion.

This may be done by admitting steam slowly and allowing it to blow through the turbine while standing idle. When started, it should be brought to speed slowly, the same as a reciprocating engine, in order to prevent any rush of water from the boiler.

The above precautions are especially important when superheated steam is used, on account of its higher temperature. When shutting down a turbine it is well to partly close the throttle before reducing the load on the generator, as it brings it under better control should there be a tendency to speed up owing to a failure of the governor to work properly.

The directions already given for the operation of the condensing apparatus apply equally well to a turbine working under the same conditions.

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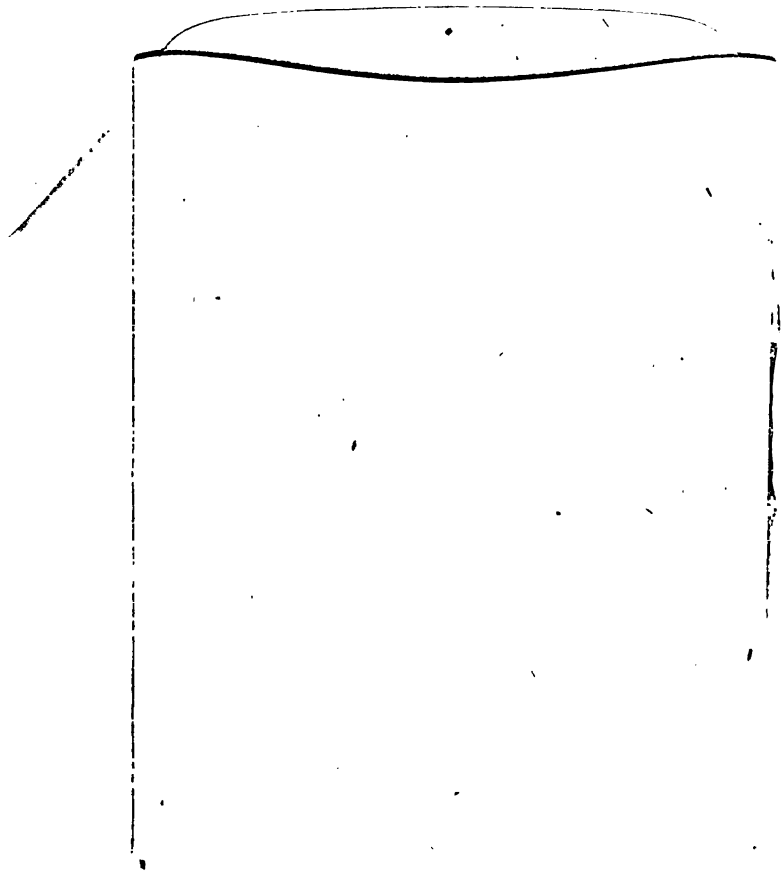

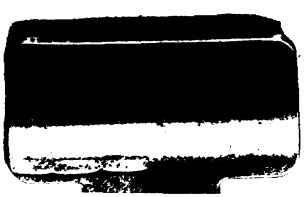

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